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JAN 76 J KELLEY, M W GOLAY DAAK02-74-C-0308

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ECONOMIC AND TECHNICAL ASPECTS OF GAS TURBINE POWER STATIONS IN TOTAL ENERGY APPLICATIONS

NUCLEAR ENGINEERING DEPARTMENT
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
CAMBRIDGE, MASSACHUSETTS

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This report summarizes a study of gas turbine power plant options which will be available in 1985 for use in total energy system applications. The gas turbines examined are of the aircraft derivative and industrial types. If coal were used as the fuel combustion chamber modification for use of low-BTU gasified coal in the power plant would be required. Such power plants could be controlled by means of the standard fuel control method, although turbine bypass control would also be possible. For total energy applications it would be necessary to replace the steam boiler of the steam bottoming cycle which is currently used in such

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applications with a high temperature water heat exchanger; however, such a design change would not be difficult to implement. Aircraft derivative turbines are more expensive but easier to maintain, transport, and repair, and are amenable to design of a more reliable power station than industrial turbines. On that basis, the former turbine type is judged to be preferable for military installation total energy applications.

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CHAPTER 1

INTRODUCTION

This report presents the results of a study of gas turbine power station options which will be available for 1985 large military installation total energy applications. In such an application a Brayton-cycle gas turbine system would be used to produce electric power, and the high-temperature exhaust gases from the turbine would be used to produce high temperature water in a thermal utility system. The temperature-entropy cycle diagram for such a system is illustrated in Fig. 1.1, and a schematic diagram of the necessary apparatus is shown in Fig. 2.2.

It is seen that the power station consists of a fuel supply system (assumed to be a low-BTU coal gasification plant), a turbine-driven air compressor, a combustion chamber in which the air and fuel react in combustion, an electric power turbine (into which flow exhaust gases from the compressor turbine) which drives an electric generator, and a high temperature water heat exchanger.

There are four major vendors of electric power-generating gas turbines - Turbo Power and Marine, Inc. (a division of United Technologies), the Westinghouse Electric Corp., the General Electric Co., and the Curtis-Wright Corp. In this study information was requested from all of these vendors, and only the former two companies cooperated. However, they are major manufacturers, respectively, of aircraft derivative and industrial type units -

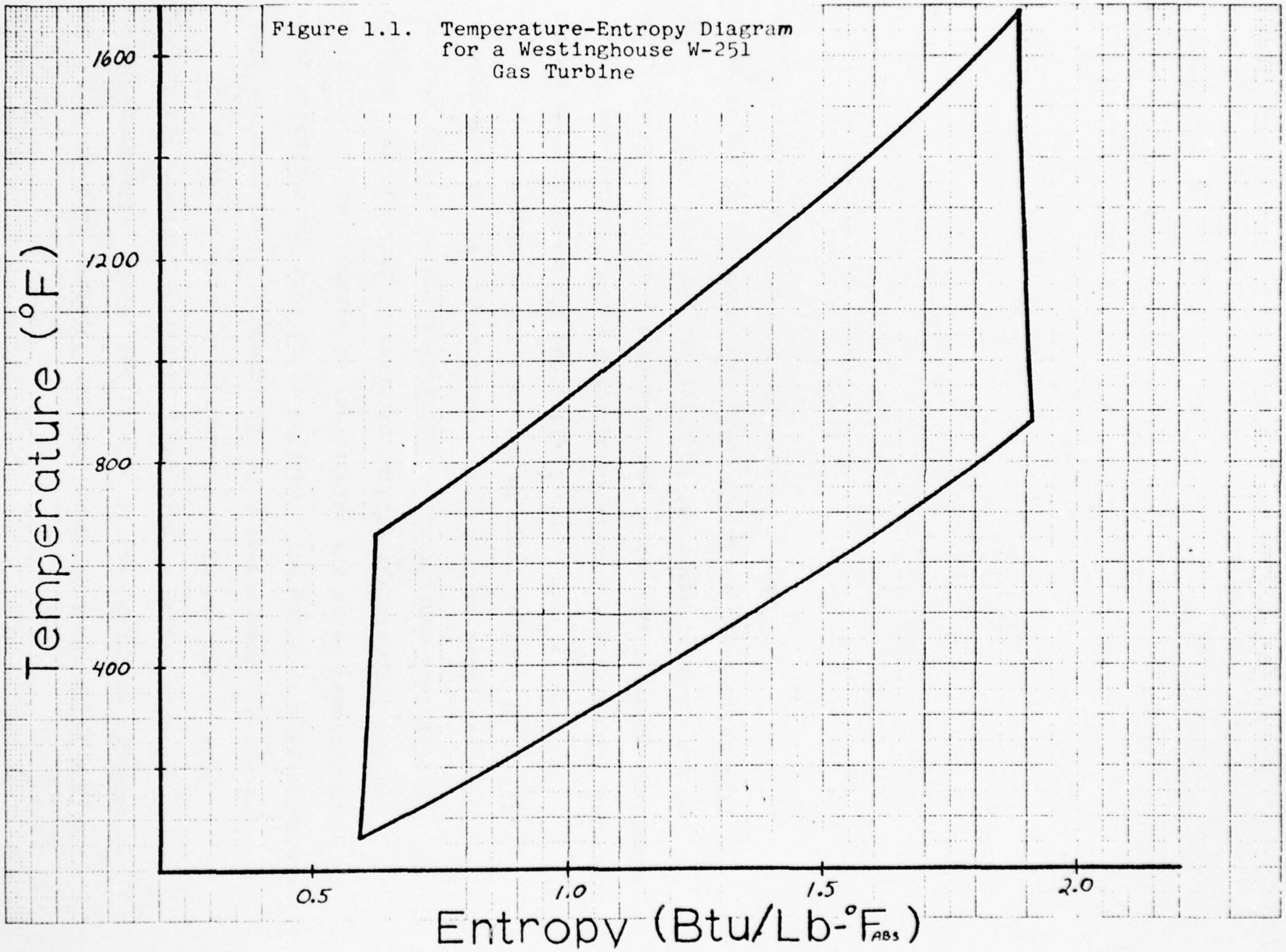


Figure 1.1. Temperature-Entropy Diagram for a Westinghouse W-251 Gas Turbine

which are the only types of gas turbine options currently available commercially. Thus, the results of the study are representative of the currently-prevailing gas turbine supply situation.

The major technical development work in the gas turbine field are concerned overwhelmingly with improvements in turbine blade materials and cooling techniques. The goals of both efforts are to permit an increase in the turbine gas inlet temperature (with a corresponding increase in turbine efficiency), and to increase turbine material lifetimes. The maximum anticipated improvements by 1985 in turbines which would be available commercially would result in efficiencies of approximately 45 to 47%, increased from 38 to 40% currently.[1]

No new major turbine development efforts are currently underway which would result in a radically new technology becoming available prior to 1985. Rather, the types of improvements which may be expected are of the minor design-perturbation type, which will result cumulatively in more efficient designs.

CHAPTER 2

DESCRIPTION OF SYSTEM OPTIONS AND REQUIRED MODIFICATIONS

2.1 System Description

A) Equipment Description

Turbo Power and Marine Systems'(TP&M) FT4C[2,3] is a jet engine derivative gas turbine. As shown in Fig. 2.1 and as detailed in schematic form in Fig. 2.2 the FT4C consists of a low pressure compressor, a high pressure compressor, a combustor, a high pressure turbine, a low pressure turbine, and a free turbine. The low pressure compressor has eight stages and is powered by the two-stage low pressure turbine on a shaft that is turning at 6425 RPM (ISO base conditions). This shaft speed is regulated and set to predetermined speeds by the control system according to the ambient temperature. The high pressure compressor (which brings the pressure ratio up to 13.3) has seven stages and is driven by the single-stage high pressure turbine on a shaft that is concentric with the "low pressure" shaft and turns at 8280 RPM (ISO base conditions). This shaft is not regulated but turns at whatever speed is the most efficient for the current aero-dynamic conditions (overspeed protection is provided). All of the electricity is generated by the free turbine, a two-stage turbine that turns at a regulated 3600 RPM.

Westinghouse's W251[4] and W501[5] are of the industrial gas turbine type, since they have a single shaft design (shown in Fig. 2.3). The W251 has a pressure ratio of 10.0 and consists of an 18-stage compressor and a 3-stage turbine on a shaft that turns at 4894 RPM. This

shaft speed, while more efficient aerodynamically (than 3600 RPM), necessitates the use of a gear box to couple the turbine to the generator which has to turn at 3600 RPM. The W501, being a larger machine, runs both a 19-stage compressor and a 4-stage turbine on a shaft turning at 3600 RPM and has a pressure ratio of 12.0.

2.2 Plant Size and Layout

TP & M offers a Power Pac (one FT4C turbine) and a Twin Pac (two FT4C turbines coupled). The plant size and layout for these two systems are given in Figs. 2.4 and 2.5. Three Twin Pacs, about 150 Megawatts of electric generating capacity, would take up an area having dimensions of 103 feet by 125 feet. Another interesting configuration that TPM offers is the Mobile Power Pac. In this system the gas turbine and generator are housed in one flat bed truck and the control house in another. Upon arrival at a site, this system can produce its full 22 MW within 4 hours. Considering the use of cargo planes as well as trucks, this unit offers an alternative to on-site redundancy. Another advantage of the TP & M design is that the gas generator internals can be replaced in as little as eight hours. Since most serious maintenance problems occur in the gas generator, the ability to replace this component so easily insures that down-time (as long as a spare gas generator is in stock) will seldom exceed eight hours.

The layouts of W251 and W501 are given in Figs. 2.6 and 2.7. There is not a significant size difference between the TPM Twin Pac (102' x 20' and 50 MW), the W251 (114' x 32' and 33 MW), and the W501 (123' x 38' and 100 MW). In an application where space

restrictions are critical, the W501 provides the greatest power per unit of area (note: the Twin Pac dimension does not include the control room).

2.3 Performance Criteria

The performance of these systems will be stated in two ways, standard (full load) performance charts for both ISO (59 °F, sea level) and NEMA (80 °F, 1000 ft. altitude) conditions, and heat rate versus load curves for ISO conditions. These data are summarized in Tables 2.1 and 2.2, and Figs. 2.8 through 2.10. Another critical factor is the time required for a system to reach full load from the shut-off condition. These times are given in Table 2.3 for two modes of startup, normal and fast for the various turbines discussed. It must be pointed out, however, that the fast mode is detrimental to component lifetime, and should only be used in emergencies. All of the performance data given in this section apply to turbines using 1000 BTU/SCF (HHV) natural gas as a fuel, since performance data are not yet available for low BTU gas operation.

2.4 System Prices

The price data quoted in this section (Table 2.4) apply to unmodified gas turbines, and they are turnkey price estimates (everything that is needed for operation of the system is included except fuel tanks). It is seen that the aircraft-derivative gas turbine power plants are substantially more expensive than the industrial turbine power plants. However, much of this price advantage is negated by the superior ease of repair, reliability, and ease of modification of the aircraft-derivative units.

2.5 Modifications for Use of Low BTU Gas

In order to run a gas turbine on low BTU gas, design changes are required in the combustion section. The major portion of the system can remain unchanged, but the manifold size and fuel nozzles must be adjusted to permit the necessary higher fuel flows (by factors of 400-900%).

Turbo Power and Marine Systems has conducted burner rig tests using a gas with a heating value of 100-150 BTU/SCF. These tests are being carried out in conjunction with Texaco Oil Co. to produce a "Gasification Power Generation System." This system accepts any commercial grade of crude or residual fuels and converts them to a low BTU gas to be used in a combined cycle plant. In line with this development, full scale engine tests will be run and a modified version of their gas turbine should be available well before 1985. Unfortunately, neither performance nor price data for this modified system are available currently.

Westinghouse, also does not have a full scale turbine running on low BTU gas. However, in the past a W-201 was used with an external combustor to burn 90 BTU/SCF gas and ran for 16,109 hours. Currently, in Japan, two turbines (a MW-171SM and a MW-301) are operating on blast furnace gas and coke oven gas and have operated for 35,000 and 27,000 hours respectively. Similar to TP & M, Westinghouse is working with Shell to develop the PACE-SGP oil gasification combined cycle plant. A modified W-251 or W-501 gas turbine should be available in a year or two.

TABLE 2.1

UNITED TECHNOLOGIES (TPM) TURBINE PARAMETERS

Performance Parameter	Operating Conditions	
	ISO	NEMA
Plant Net Power (KW)	26,300	22,600
Plant Heat Rate (BTU/KWH)	11,500	11,800
Exhaust Flow (lb/hr)	1,040,400	900,000
Exhaust Temp. (°F)	829	842
Fuel Flow (SCF/hr)	336,056	296,311
Inlet ΔP (in. of H_2O)	4.0	4.0
Exhaust ΔP (in. of H_2O)	1.0	1.0

TABLE 2.2

WESTINGHOUSE TURBINE PERFORMANCE PARAMETERS

Performance Parameter	W-251		W-501	
	ISO	NEMA	ISO	NEMA
Plant Net Power (KW)	33,619	29,822	97,820	86,.
Plant Heat Rate (BTU/KWH)	12,080	12,449	10,590	10,830
Exhaust Flow (lb/hr)	1,273,800	1,181,700	2,815,000	2,575,000
Exhaust Temp. (°F)	880	892	990	1,004
Inlet ΔP (in. of H_2O)	2.4	2.4	0.0	0.0
Exhaust ΔP (in. of H_2O)	1.6	1.6	0.0	0.0
Fuel Flow (SCF/hr)	451,242	412,505	1,151,015	1,043,771

TABLE 2.3. START UP TIMES FOR VARIOUS TURBINES

	TP&M		W251		W501	
Procedure	Normal	Fast	Normal	Fast	Normal	Fast
Time to Synchronous Speed (minutes)	3	2	12	10	15	15
Time to Load (minutes)	1/2	1/2	10	2	15	5
Total Startup Time (minutes)	3 1/2	2 1/2	22	12	30	20

TABLE 2.4. GAS TURBINE POWER PLANT PRICE DATA*

Vendor	Turbine-Unit	Power Capacity (MWe)	Price (dollar)	Specific Price (dollars/KW-capacity)
TP&M	FT4C Power Pac	26.3	3,900,000	148
TP&M	FT4C Twin Pac	52.6	6,000,000	125
Westing-house	W251	33.9	3,644,000	108
Westing-house	W501	97.8	8,139,000	83

*Stated in 1975 dollars.

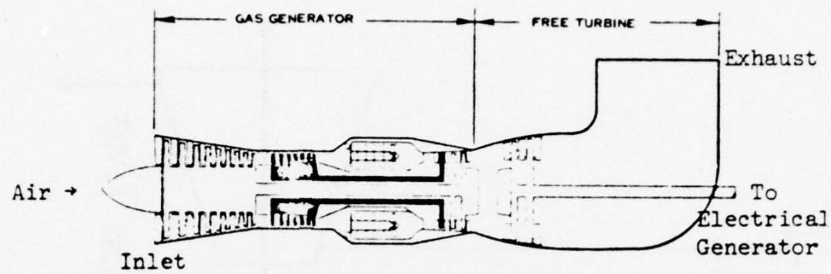


Figure 2.1.A. Complete Power Plant

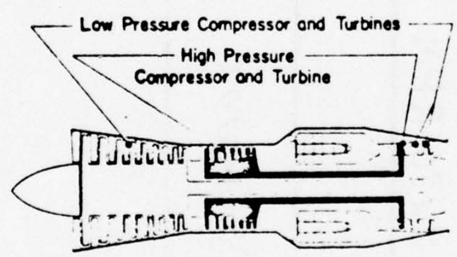


Figure 2.1.B. Detailed View of Gas Generator Section

Figure 2.1. Turbo Power and Marine FT4C Gas Turbine

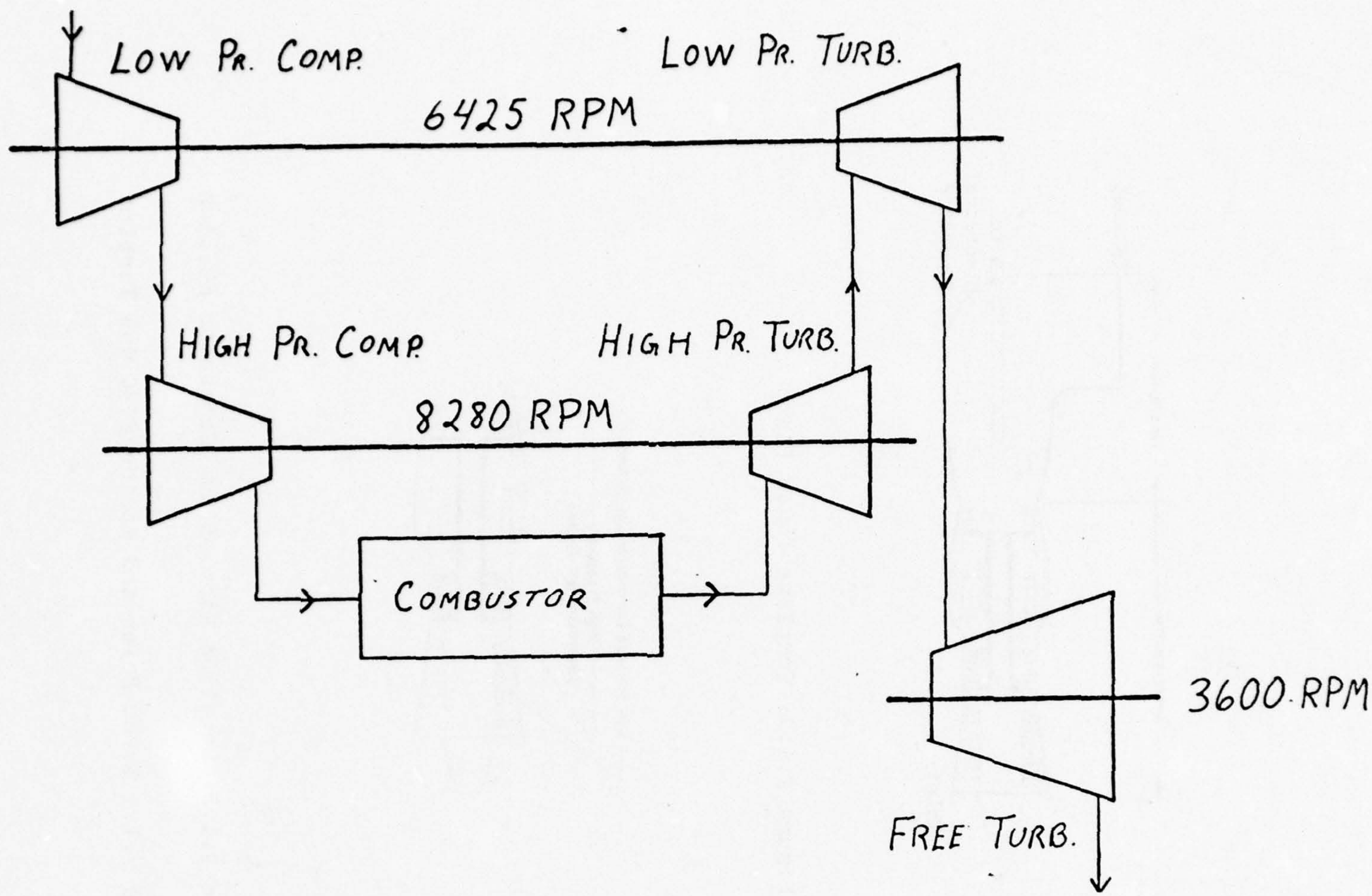
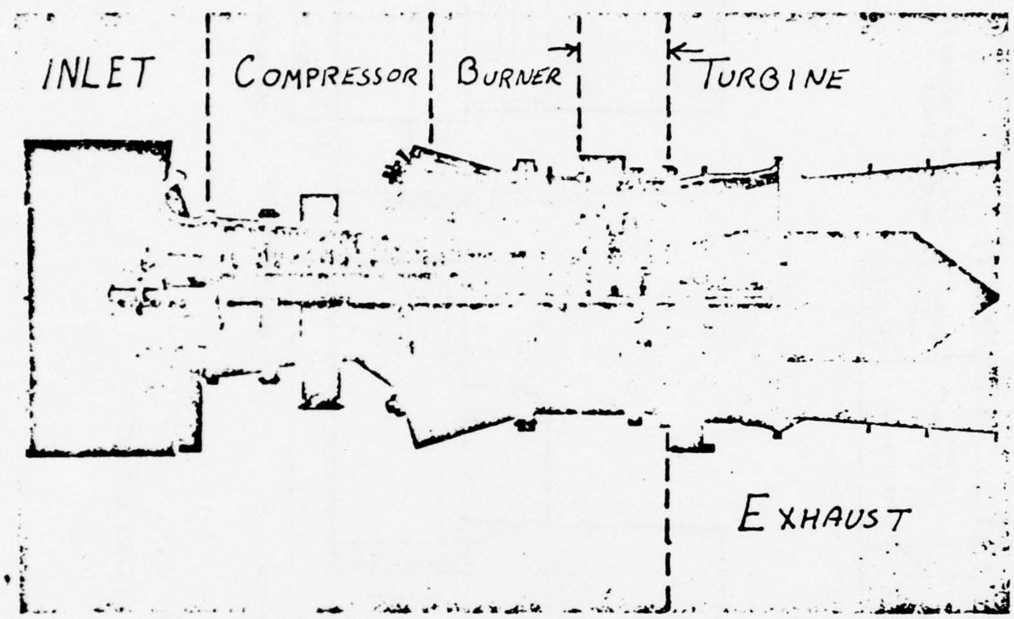


Figure 2.2. Schematic Diagram of TP&M FT4C Gas Turbine



W-251 Engine

Figure 2.3. Westinghouse W-251 Gas Turbine

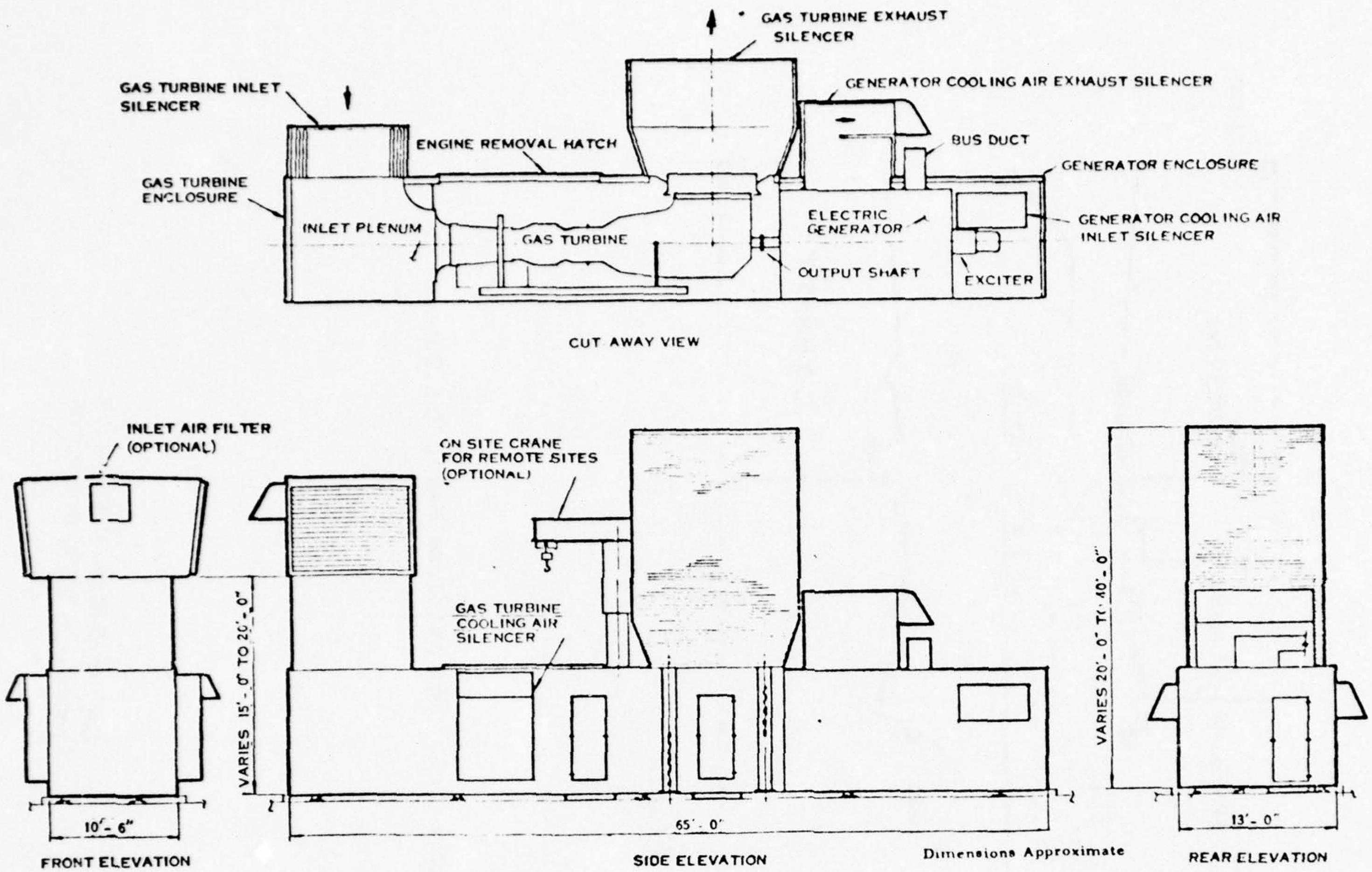


Figure 2.4.

STANDARD FT4 POWER PAC

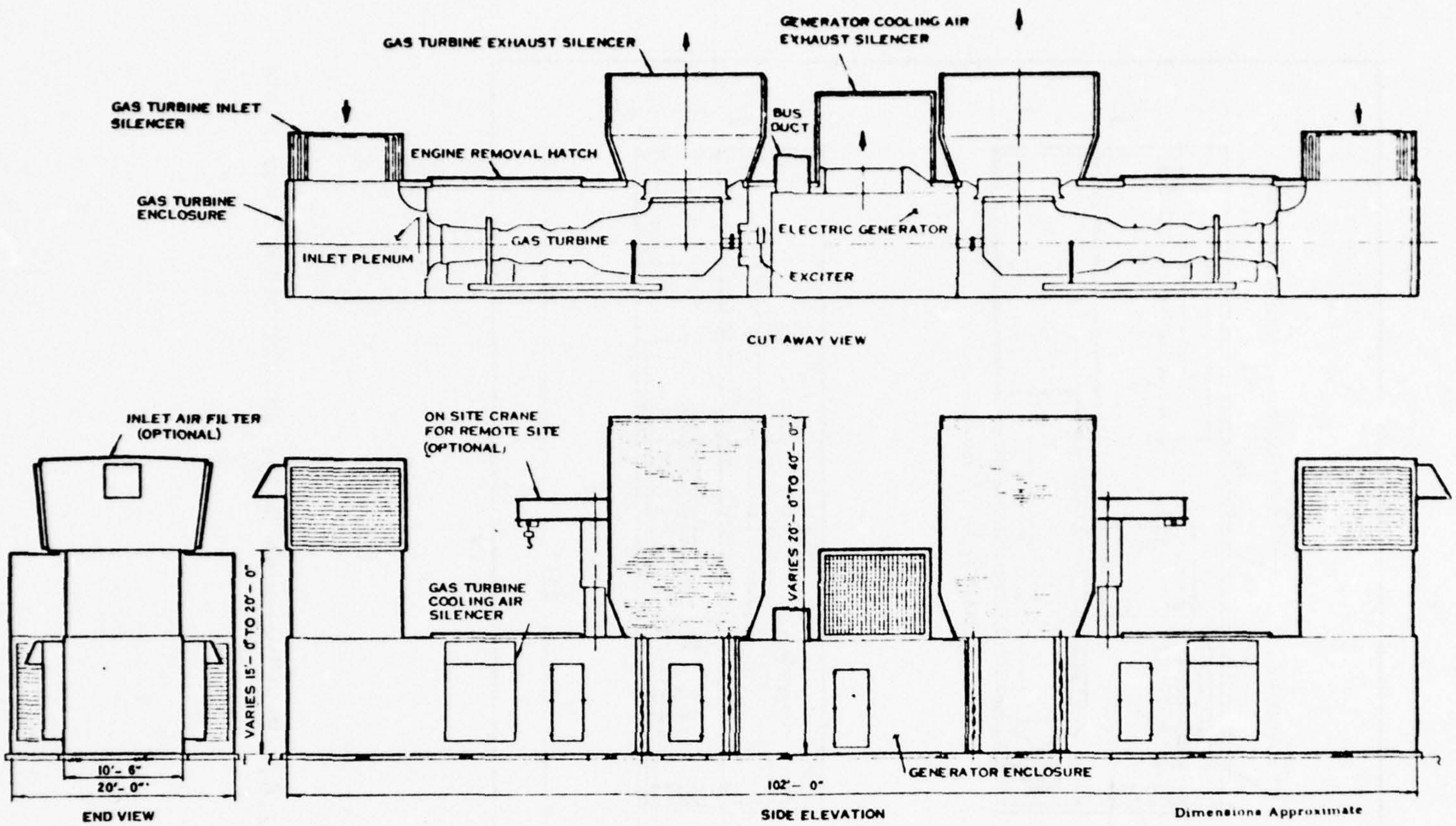


Figure 2.5.
STANDARD TP4-2 TWIN PAC

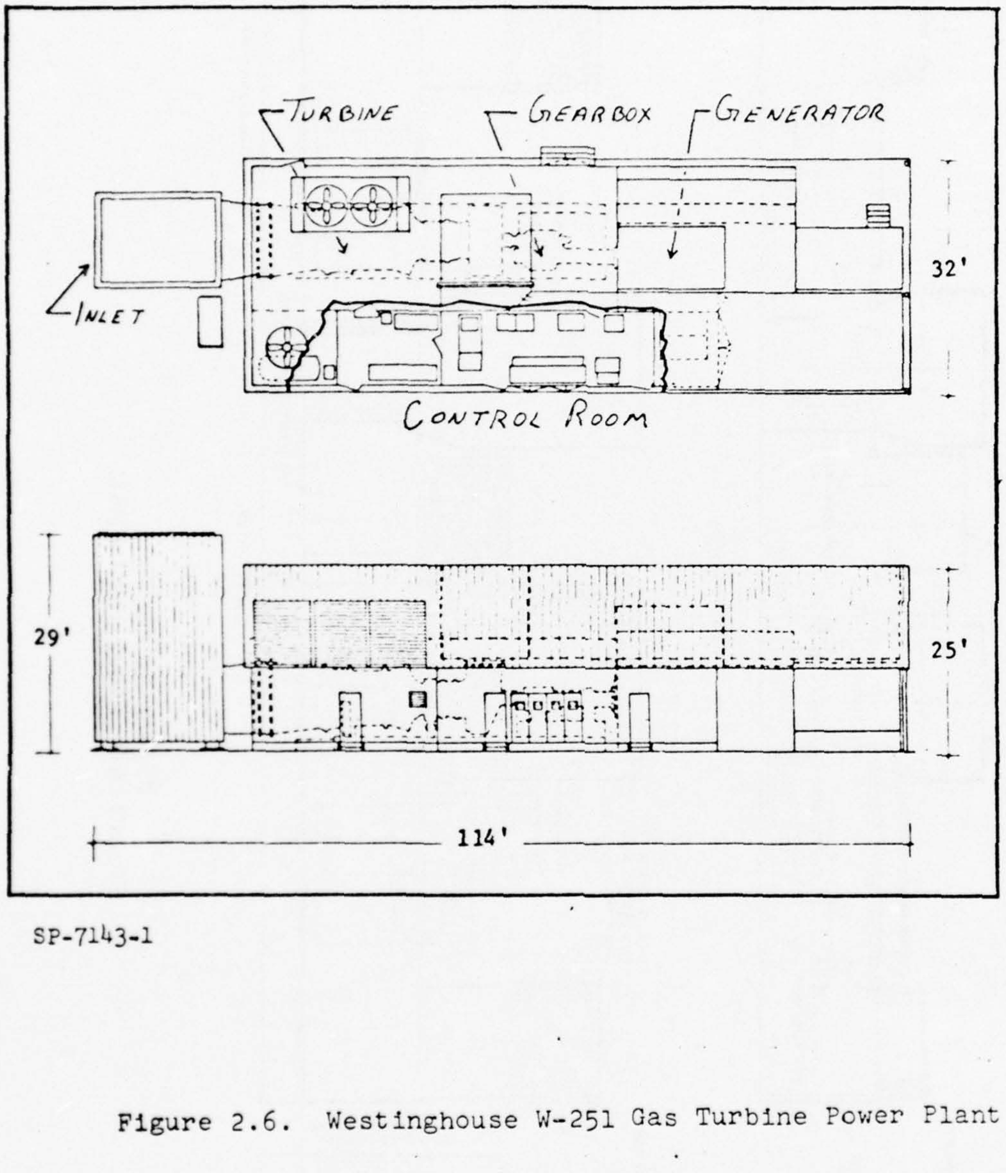


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Figure 2.6. Westinghouse W-251 Gas Turbine Power Plant

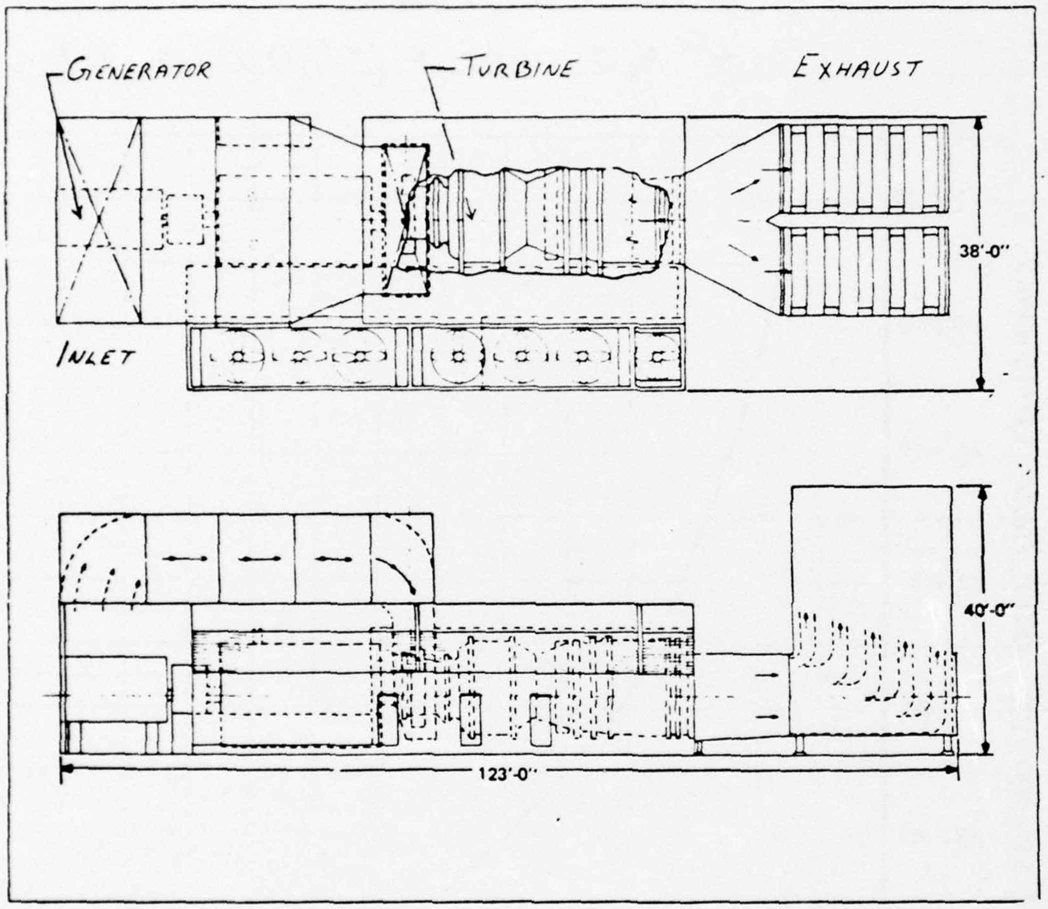


Figure 2.7. Westinghouse W-501 Gas Turbine Power Plant

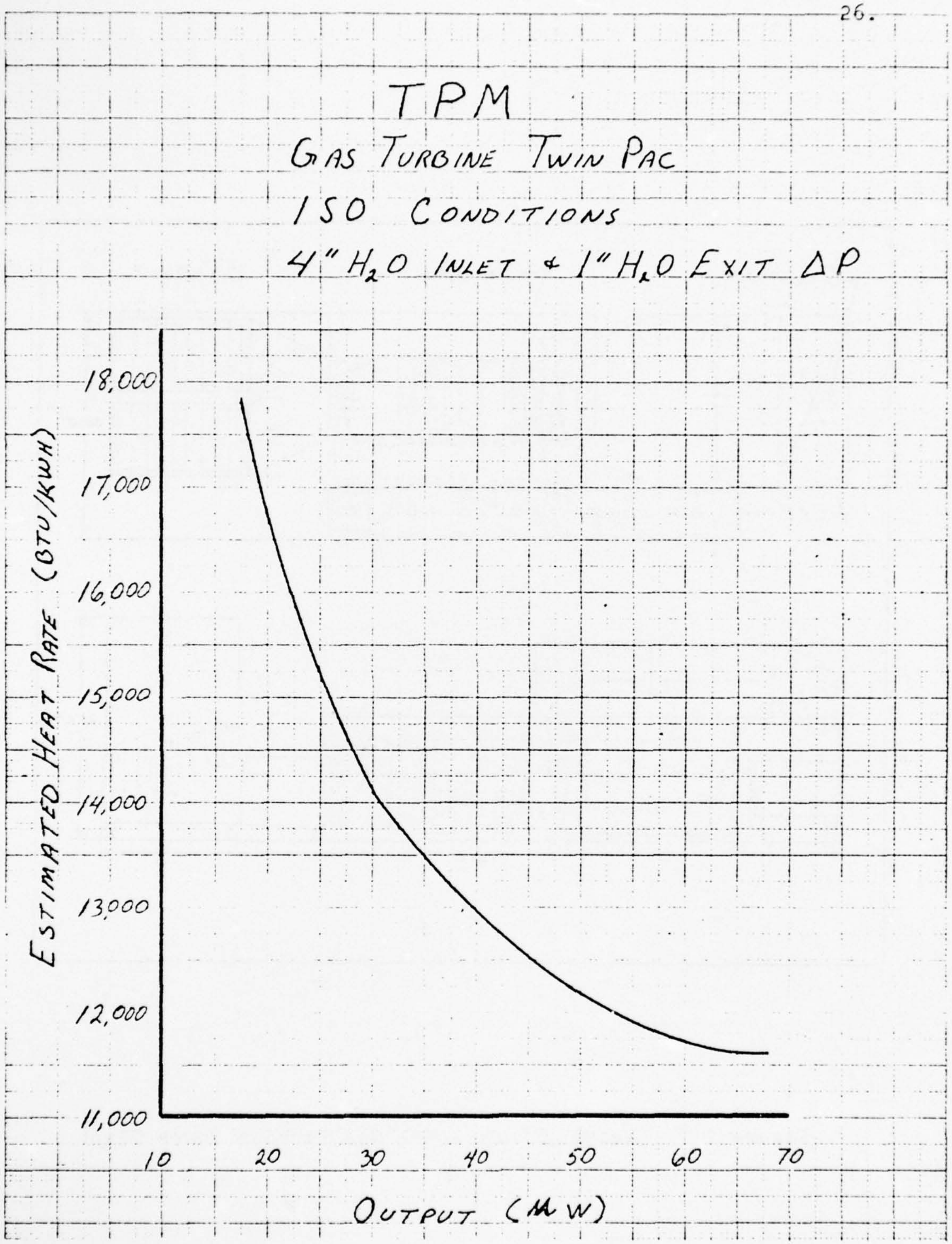


Figure 2.8. TP&M FT4C Twin Pac Heat Rate vs. Output Curve

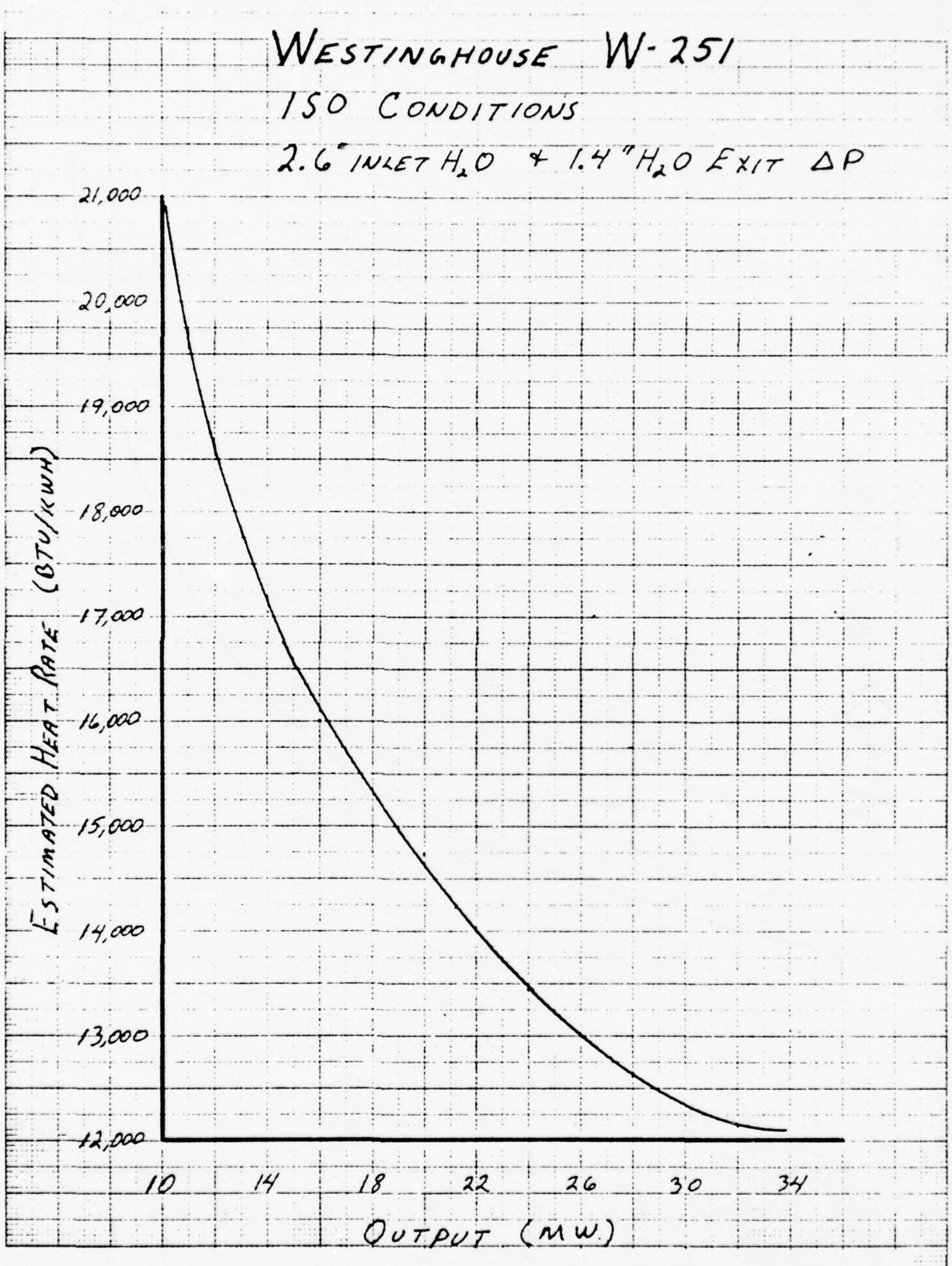
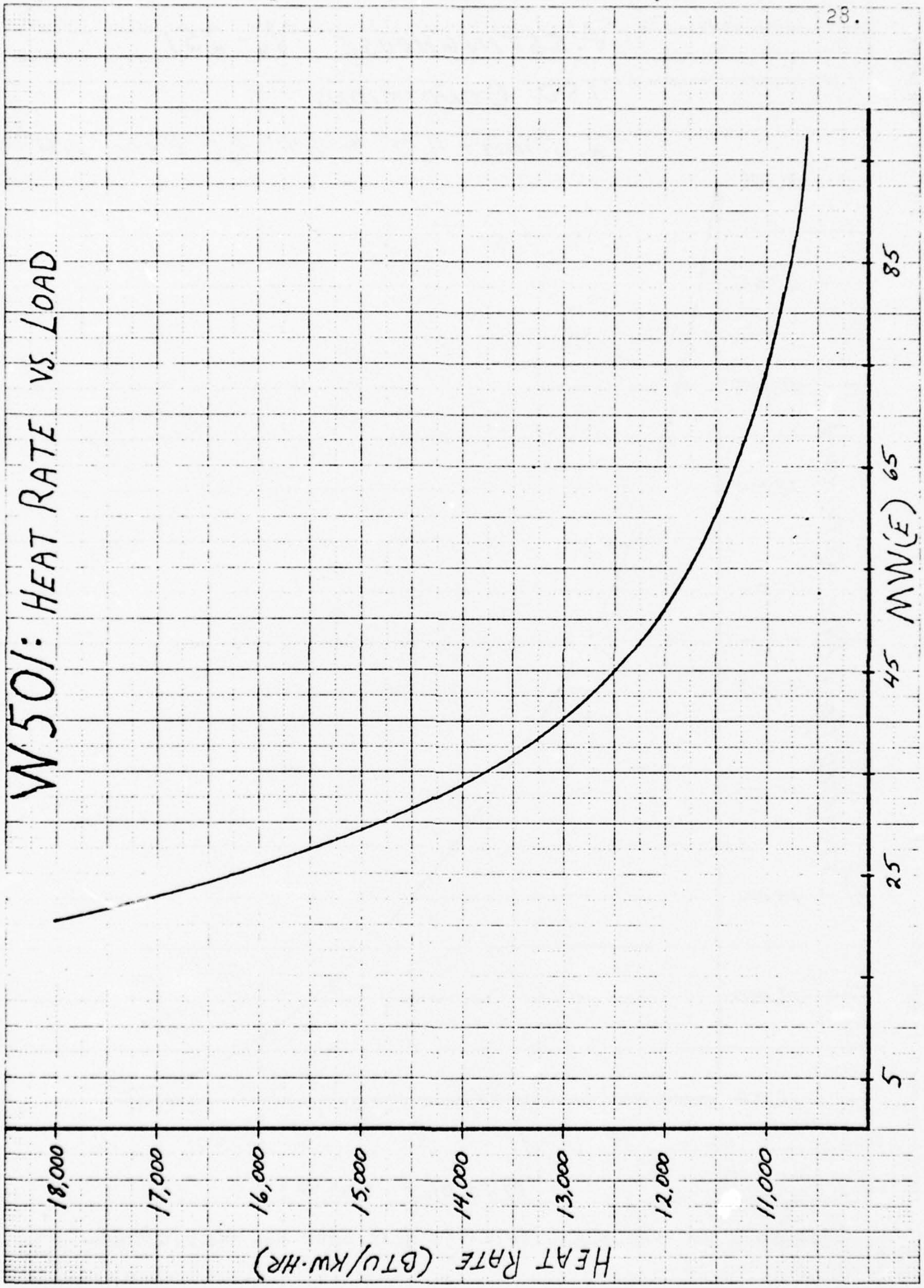


Figure 2.9. Westinghouse W-251 Heat Rate vs. Output Curve

Figure 2.10. Westinghouse W-501 Heat Rate vs. Output Curve



CHAPTER 3

CONTROL SCHEMES

3.1 Fuel Flow Control

Gas turbines currently employ this method of control due to its simplicity. When the fuel supply is decreased, the enthalpy of the gases entering the turbine is lowered, less power is produced and hence the compressor must consume less power. Therefore, less air gets compressed and the gas turbine returns to steady state operation with a lower mass flowrate and power output. This method is inherently stable and very easily instrumented, unfortunately, it can only be used to control the electrical load. A discussion of the fuel flow control scheme for each manufacturer follows.

TP&M uses an electronic fuel control[6] that serves three basic functions:

1) governing the speed of the gas generator and free turbine,
2) limiting acceleration to prevent an over-temperature condition, and
3) shutting down the engine when certain limits are exceeded.

The system controls operation of the engine by positioning a fuel modulating valve through the three basic control loops of high pressure compressor-turbine speed (modified by ambient temperature), free turbine speed, and free turbine inlet gas temperature.

The first control loop regulates the speed of the high pressure compressor and turbine shaft. Design speeds for ISO and NEMA conditions are 8330 and 8385 RPM, respectively. For other conditions this speed is automatically set to insure good performance and maximum engine life. The speed of the free turbine can be set to

any speed or to one of two fixed reference speeds (3000 RPM for 50 Hz and 3600 RPM for 60 Hz electrical power). This loop also includes a power sensor to stabilize the engine control by preventing cycling of the fuel control valve, and an overspeed trip that will shut down the engine if overspeed occurs. The third loop controls the inlet temperature of the gases entering the free turbine to 1175 °F. Whenever this limit is reached, engine output will be limited, thus protecting the engine against long periods of operation at elevated temperatures. During off-peak conditions this temperature is maintained so that maximum efficiency will be achieved. The signals from these three control loops are fed to a logic circuit which selects the signal that calls for the least fuel flow.

Westinghouse uses a hybrid analog/digital system[4] which provides centralized automatic speed and temperature control for its gas turbines at all stages of operation, from no load to maximum load capability. Rather than using an analog control in which the smallest signal controls the fuel flow (as with TP&M), the digital part of Westinghouse's control system calculates the appropriate fuel flow. Input data to this calculation are a reference speed, turbine speed, electrical power output, combustor shell pressure, and temperatures at various points in the system. Provisions are made for maintaining cycle temperatures, turbine-generator speed, load and loading rates, for controlling acceleration during starting for bringing the turbine to full speed within a fixed interval of

time and for maintaining compressor surge margin. The times from start-up to full load are given in Table 2.3 for both manufacturers.

Figures 3.1 through 3.3 show differing trends in the thermal-electric power generation ratios for the Westinghouse turbines as contrasted to the TP&M turbine. These differences are due to the gas generator concept used in the aircraft derivative turbine. In the FT4C, as the fuel and air mass flowrates are decreased together, the free turbine inlet temperature is held constant and the rotating speeds of the two shafts in the gas generator are decreased. Thus, the high efficiency of the gas generator is maintained even though the flowrate changes by as much as 50%. Due to generator constraints, the speed of the free turbine is maintained at 3600 RPM and hence the lower mass flowrate is accompanied by a decrease in the free turbine efficiency, which is reflected in an increase in the exit temperature of the turbine. However, the gas flowrate will have been decreased so much that the thermal output of the HTW heat exchange will decrease with decreasing electrical output.

In the Westinghouse systems, the compressor-turbine inlet temperature is maintained constant, and the gas flowrate is decreased. In this case, the lower mass flowrate affects both the compressor and free-turbines, as well as the compressor resulting in a significant decrease in the efficiency of these components. Consequently, the point at which no electricity is generated and at which the turbine is able to drive only the compressor is reached at 92% of the design gas flowrate. Also, due to the rapid decrease in turbine efficiency, as the gas flowrate decreases the outlet temperature of

the turbine will have increased. Since the gas flowrate will have decreased only marginally, the thermal output will have increased while the electrical output will have been lowered. This trend is reflected in a steeper heat rate curve for Westinghouse turbine than for the TP&M unit.

3.2 Turbine Bypass Control

In this scheme, hot gases are extracted before they enter the turbine and are combined with the turbine exhaust to produce a hotter exhaust system. This stream can then transfer more heat to the waste heat recovery unit, making it possible to tailor the output to both the thermal and electrical loads. Due to the intricate construction of these turbines, gases cannot be extracted between turbine stages without a significant design change. The result of this is that for the Westinghouse turbines, gases can only be extracted from the exhaust of the combustion chamber at full load temperatures of 1870° and 2000 °F for the W-251 and the W-501, respectively. For the TPM turbines, gases can be extracted from either the combustion chamber exhaust or from the free turbine inlet, at full load temperatures of 1825° and 1175 °F. For all of these turbines, significant changes in the construction of the combustion chamber would have to be made to permit the turbine bypass, whereas for a free turbine bypass in the TPM design only a relatively minor adaption must be made. Also, due to the "jet engine" type design of TPM, where the first two turbines operate the compressors, it is impossible to extract more than a few percent of the total flow and hence it is impractical to do so.

When a turbine bypass control scheme is used, two major consequences result:

1) Less energy is delivered to the turbine, and since the compressor load remains unchanged, the proportion of power developed by the turbine that is used to run the compressor increases and net power is reduced, and

2) the mass flowrate through the turbine is decreased, causing a degradation of the aerodynamic performance (eg. a 10% change in the mass flowrate can decrease the efficiency of the turbine from 90 to 84%).

This means that less electricity is generated than a simple energy balance would predict and certain practical limits upon the thermal to electric power ratio are imposed by the nature of the system. Figures 3.1 through 3.3 show the electric and thermal power produced when the turbine is running at full capacity.* They were calculated using the "off-peak" performance estimation method[7] which is explained in Appendix A. The heat outputs are calculated using the assumption of an exit temperature from the waste heat recovery unit of 343 °F which is temperature used in the design of this unit for a TP&M Power-Pac in Chapter 4.

The range of thermal and electric demands that can be met using this control method is remarkably similar for all three of the turbines studied. Only one major difference is apparent, that being the much larger range of the allowable bypass flow for the TP&M turbine (55%) compared to the Westinghouse turbines (11 and 14%). This difference is a direct consequence of the "aircraft-derivative" design of TP&M's FT4 turbine. In this system the free turbine is

*And using 1000 BTU/SCF natural gas.

decoupled from the gas generator and hence, bypassing the free turbine has no effect on the turbines driving the compressors. Therefore, aerodynamic performance degradation occurs only in the free turbine and not in the two turbines of the gas generator. The free turbine concept of the FT4C gives it three advantages over industrial type gas turbines in a system utilizing a turbine bypass control scheme:

1) Extracting the exhaust gases before the free turbine necessitates only a minor ducting modification whereas in the industrial type turbine the entire casing surrounding the combustion chamber must be redesigned,
2) The temperature of the extracted gases is much lower, 1175 °F compared to 1875 °F and 2000 °F, and
3) the larger bypass flowrate range makes the system performance less sensitive to the bypass flow valve and therefore easier to control.

3.3 Outlet Pressure Control

As far as pressure is concerned, the compressor can be considered to be isolated from the turbine. Hence, if the turbine is operating against a back-pressure, the compressor will continue to deliver air to the combustion chamber at the design pressure ratio (as long as enough shaft power is supplied to the compressor). Meanwhile, due to the increased back pressure, the pressure of the turbine exhaust is higher than its design value but the turbine inlet pressure remains the same as its design value. This causes the gases

in the turbine to expand through a lower than usual pressure ratio. Consequently, the exhaust gases have a higher pressure and temperature than in the design states and less power is developed by the turbine.[8,9] Once again, this has the dual effect of lowering electrical output and raising the thermal output. Estimated performance for the three turbines in this operational mode is presented in Figs. 3.4 to 3.6. The range of electrical to thermal loads that can be met is similar to that of the turbine bypass control scheme. Outlet pressure control does have an advantage in that only minor ducting changes and installation of a "butterfly" type valve are necessary for all three turbines. Also, the large range of back-pressure between minimum and maximum loads allows easier, less sensitive control of system.

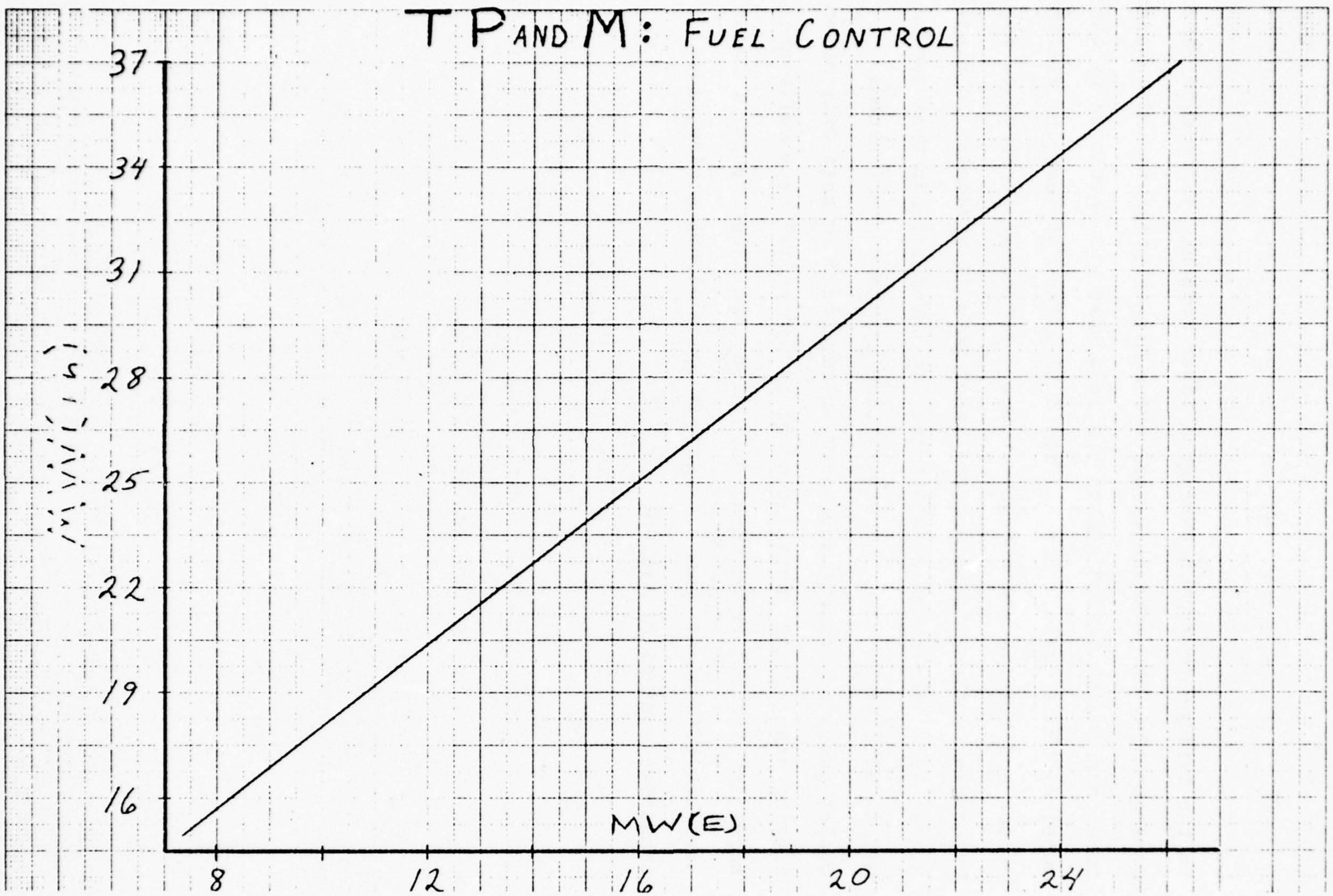


Figure 3.1. Thermal-Electrical Product Mix for Fuel Controlled TP&M FT4C Gas Turbine

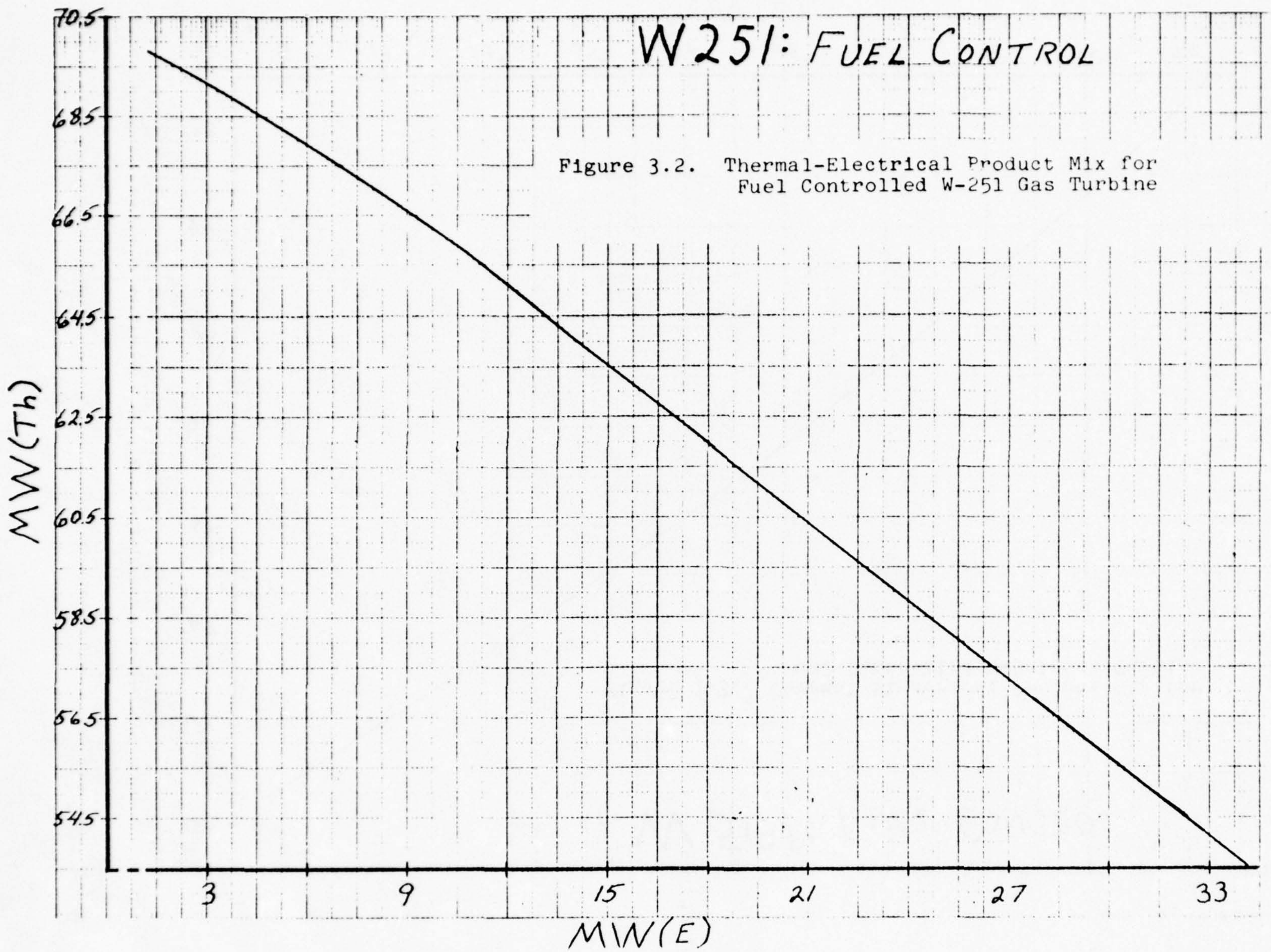


Figure 3.2. Thermal-Electrical Product Mix for Fuel Controlled W-251 Gas Turbine

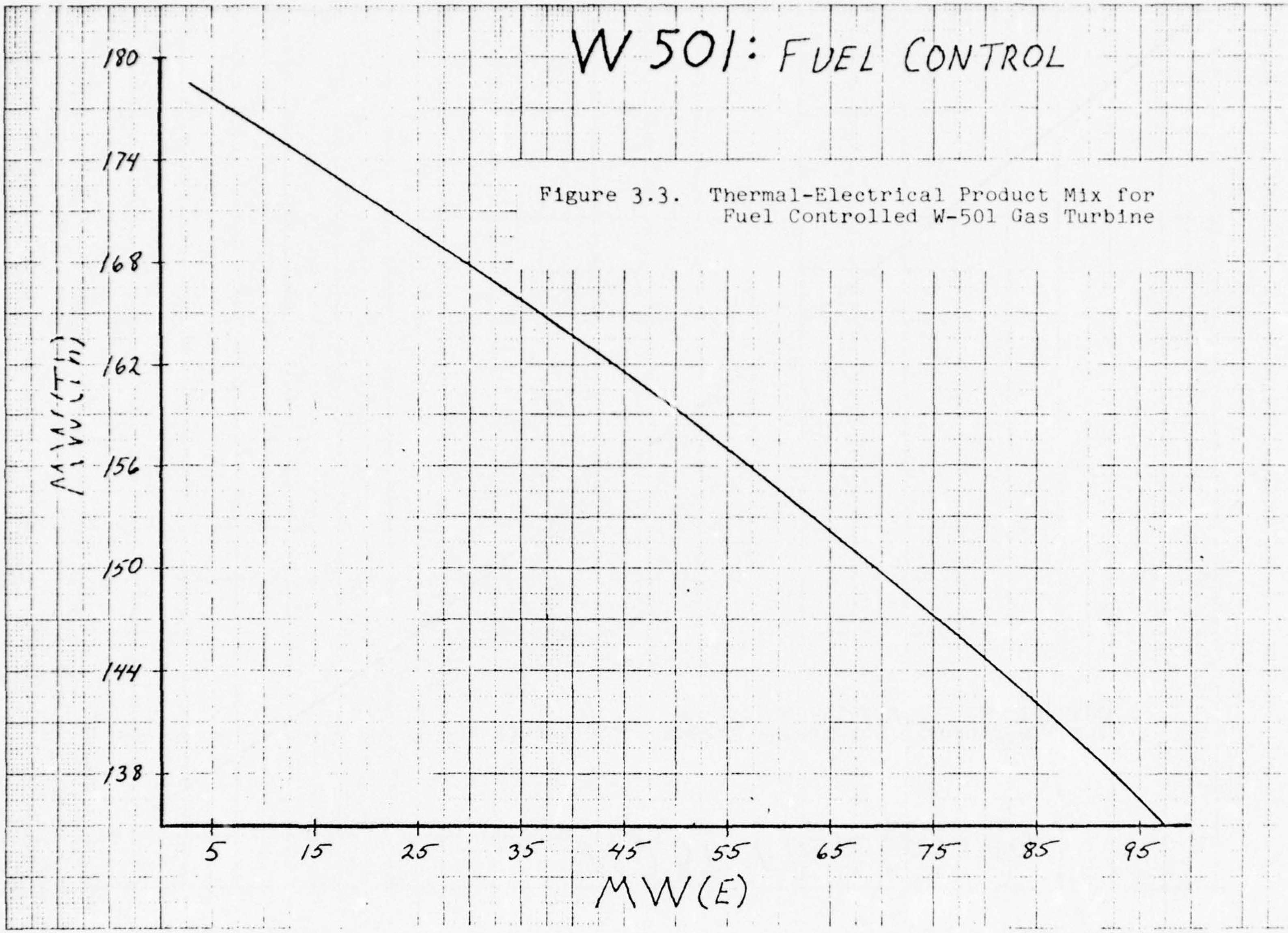


Figure 3.3. Thermal-Electrical Product Mix for Fuel Controlled W-501 Gas Turbine

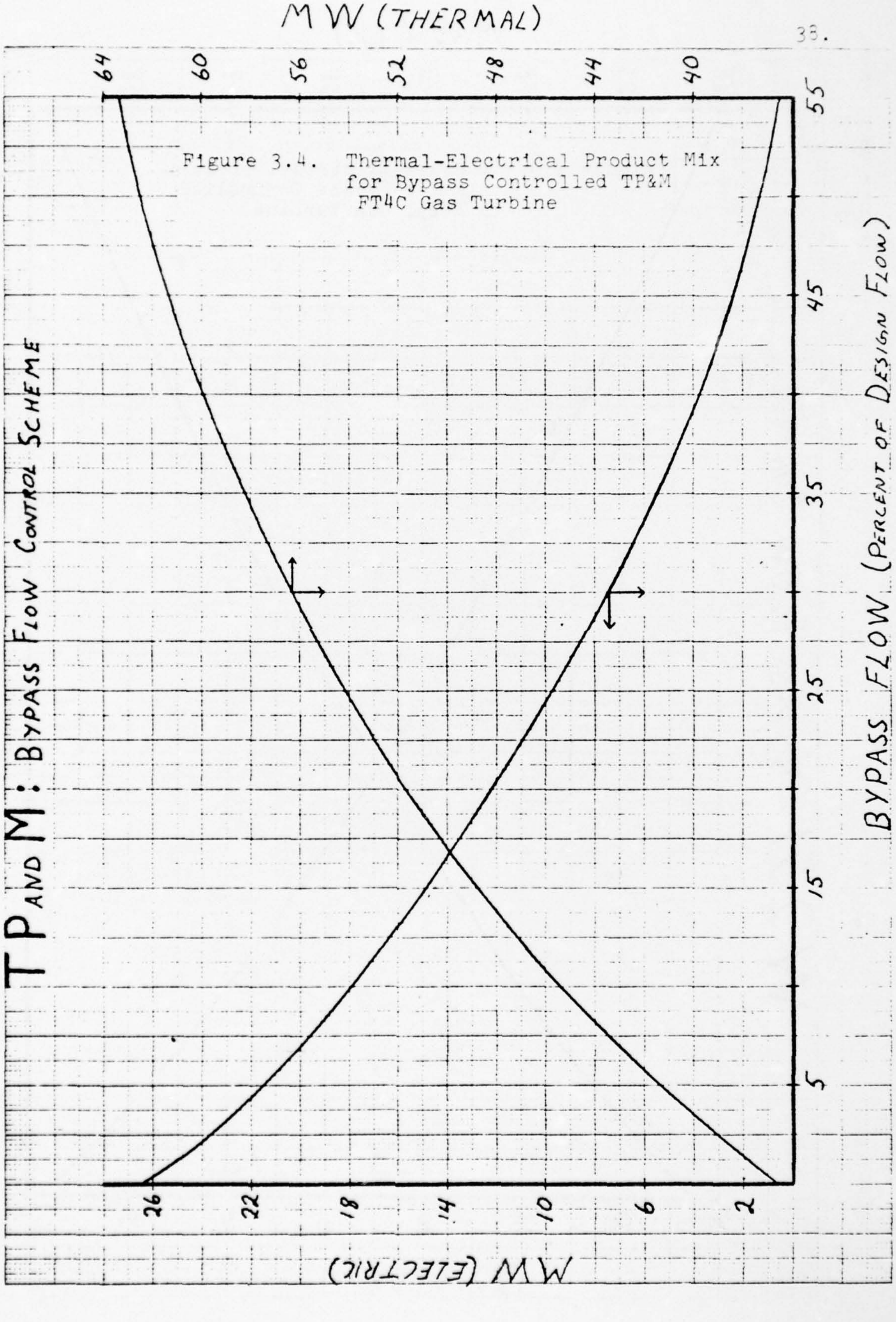


Figure 3.4. Thermal-Electrical Product Mix for Bypass Controlled TP&M FT4C Gas Turbine

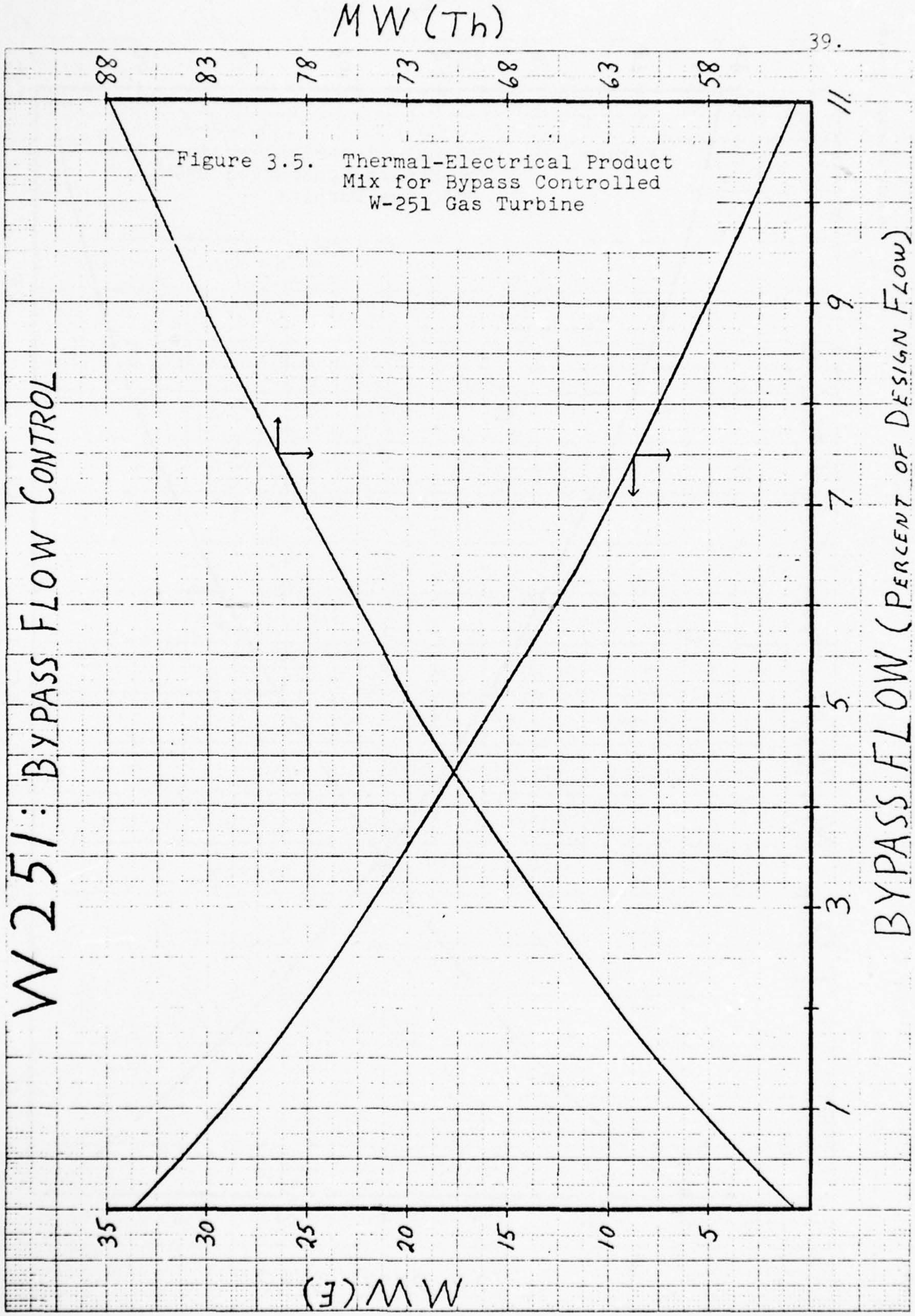


Figure 3.5. Thermal-Electrical Product Mix for Bypass Controlled W-251 Gas Turbine

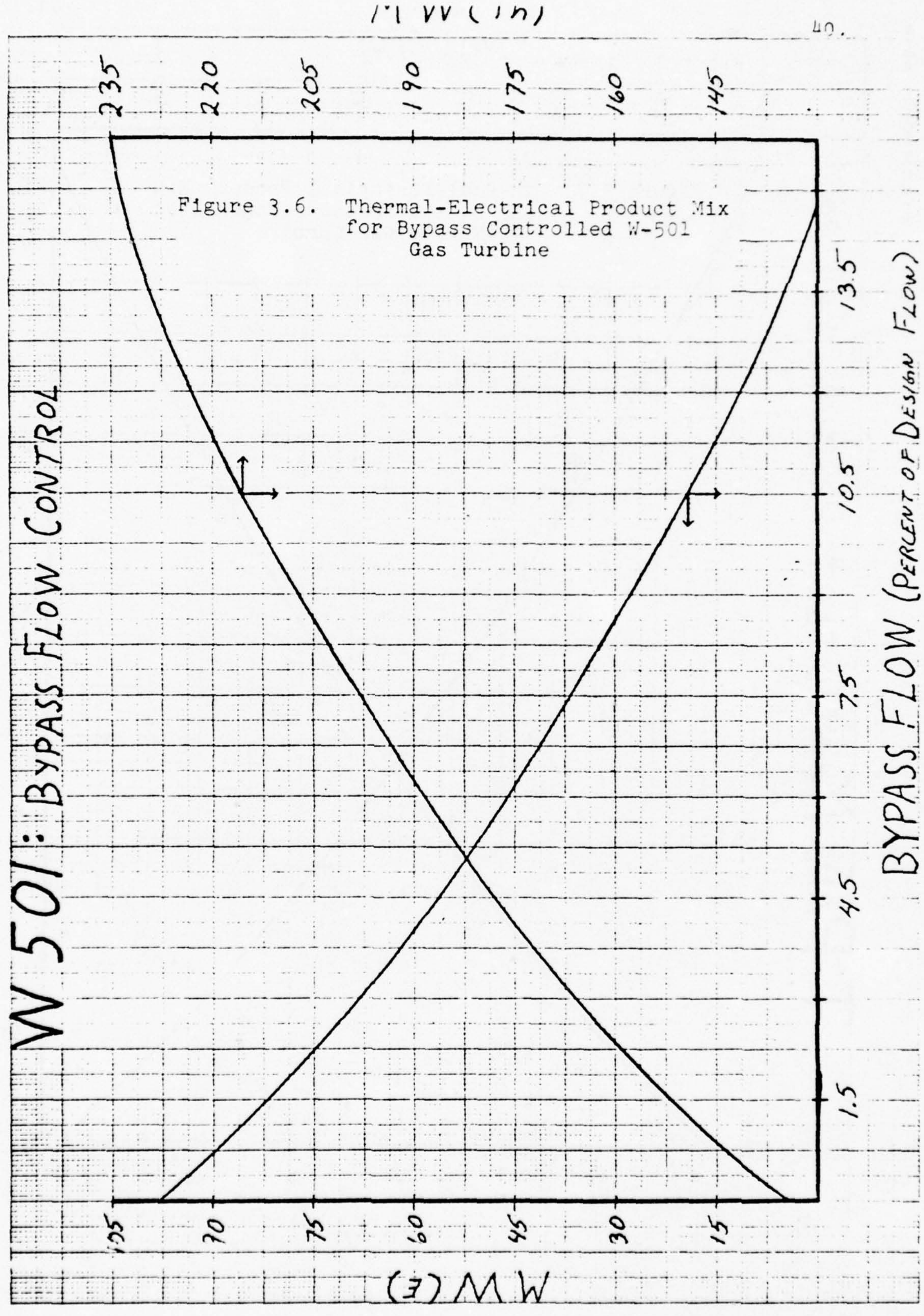


Figure 3.6. Thermal-Electrical Product Mix for Bypass Controlled W-501 Gas Turbine

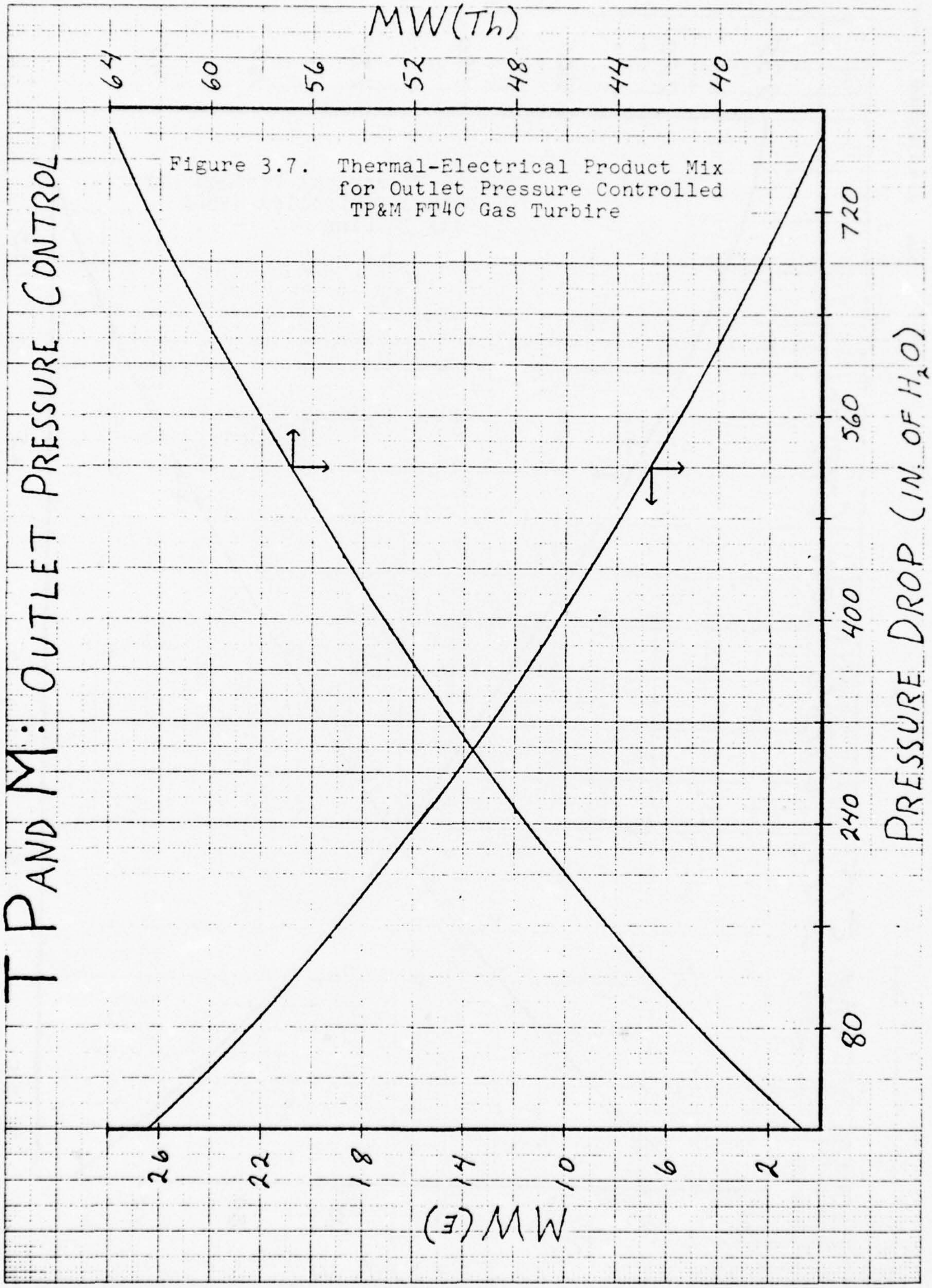


Figure 3.7. Thermal-Electrical Product Mix for Outlet Pressure Controlled TP&M FT4C Gas Turbire

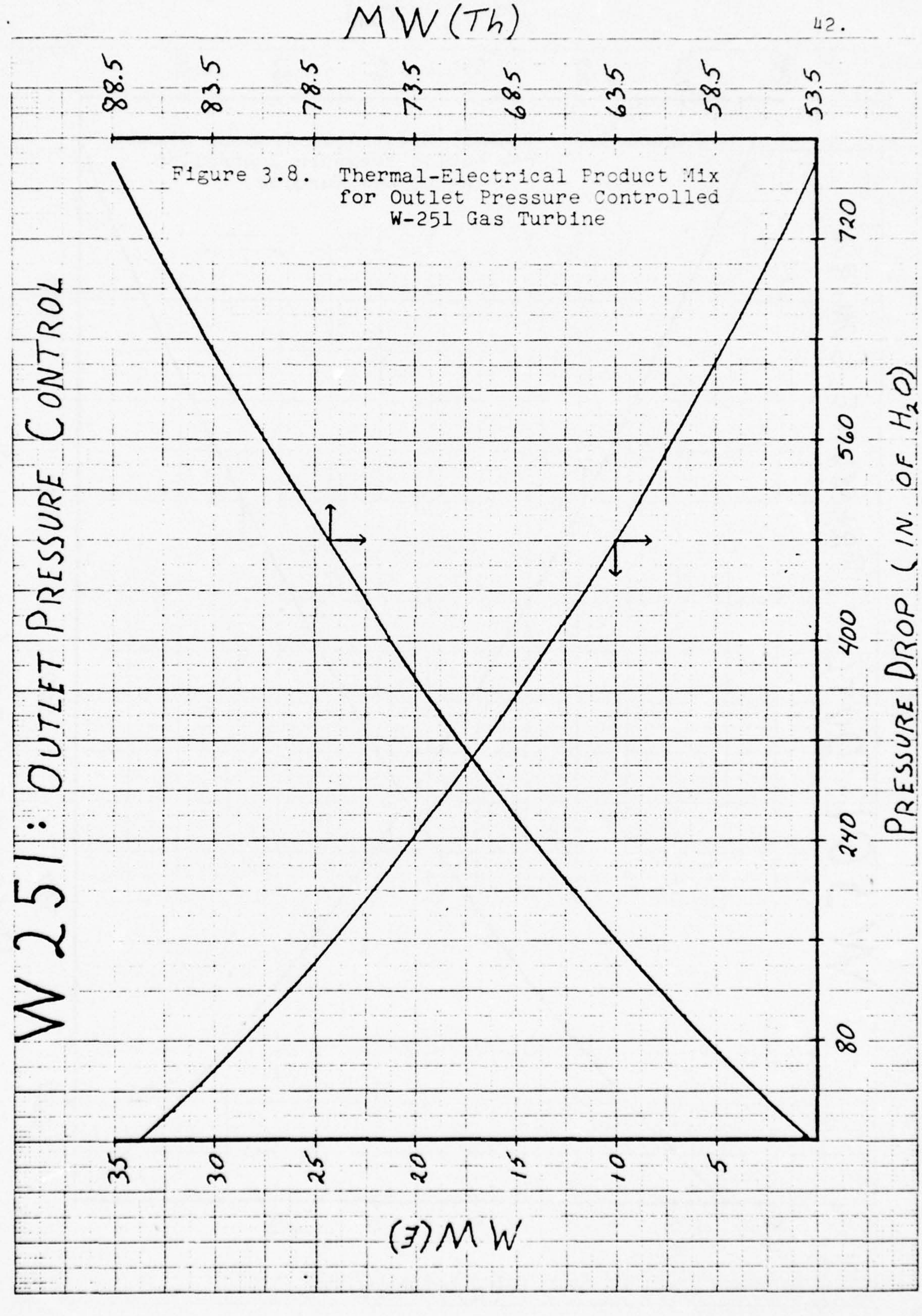


Figure 3.8. Thermal-Electrical Product Mix for Outlet Pressure Controlled W-251 Gas Turbine

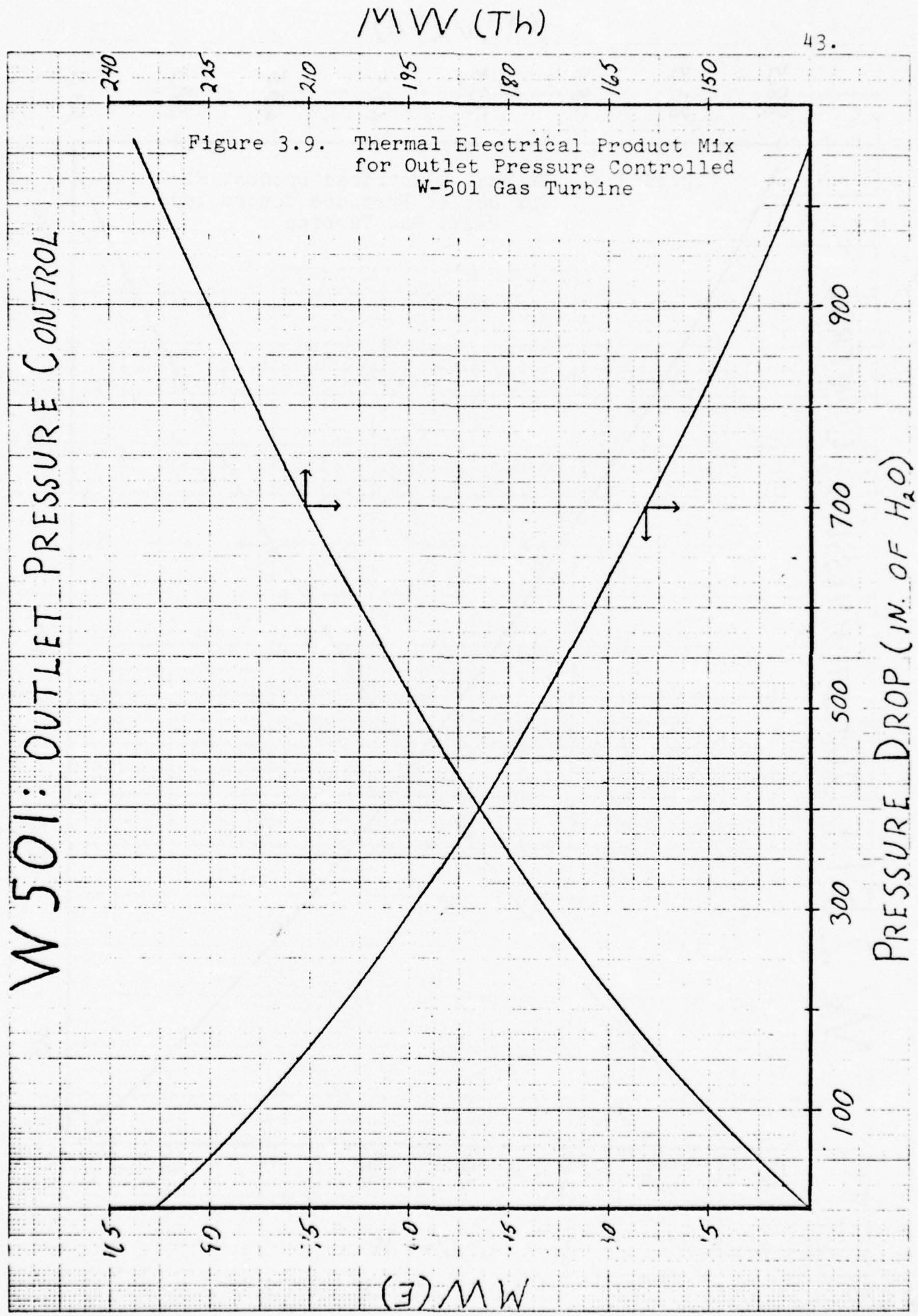


Figure 3.9. Thermal Electrical Product Mix for Outlet Pressure Controlled W-501 Gas Turbine

CHAPTER 4

THERMAL-ELECTRICAL LOAD TAILORING

As examples of the types of load-tailoring situations which could be encountered in using a gas turbine power plant to drive a total energy system for a large (50,000 population) military base the following cases are considered:

1) The hottest summer day (with 75% absorptive air conditioning),

2) The coldest winter day, and

3) A typical spring day (with no space heating).*

The resulting thermal and electrical loads[10] are summarized in Table 4.1.

The largest electrical load that needs to be met is 140 MW (occuring on the peak winter day). Four systems which can meet this demand are summarized in Table 4.2 along with pertinent data. In this chapter and in the next the merits of these systems are examined in terms of a variety of criteria. The abilities of these turbines to meet varying electrical and thermal loads are discussed in Chapter 3. In the cases considered only during one time period (Winter, 23:00-07:00) is more heat required for the Thermal Utility System than is generated normally by the turbines when meeting the electrical load. Estimates of fuel consumed during successive 8-hour intervals in the days examined are summarized in Table 4.3. It is seen that the greatest and most uniform

*Loads determined for Ft. FESA[11] using TDIST,[12]

TABLE 4.1. EXAMPLES OF SEASONAL LOADS FOR FT. FESA

	Electric Load (MW)	Thermal Load (MW)	Air Temperature (°F)
SUMMER (75% absorptive air conditioning)			
07:00-15:00	95	44	85
15:00-23:00	88	49	87
23:00-07:00	51	18	73
WINTER			
07:00-15:00	132	204	13
15:00-23:00	112	163	24
23:00-07:00	108	214	6
SPRING (no space heating)			
07:00-15:00	75	--	60
15:00-23:00	71	--	65
23:00-07:00	42	--	50

TABLE 4.2. 140 MW (ELECTRIC) SYSTEMS POWER STATION CONFIGURATION

System	Full Load Electric Power (KW)	Waste Heat at Full Electric Load (KW)	Heat Rate at Full Load (BTU/KW-HR)	Approximate Turnkey Cost (million $)
Three TP & M Twinpacs	140,100	220,080	11,600	19.80
Five W-251's	168,095	268,740	12,080	18.22
Two W-501's	195,640	269,534	10,590	16.28
One W-501 and Two W-251's	165,058	242,263	11,197	15.43

TABLE 4.3. FUEL USAGE VS. SEASONAL LOADS

SYSTEM Fuel Usage (MM SCF)	SUMMER			WINTER			SPRING			Average Fuel Usage Over These Three Days (MM SCF)
	07:00 to 15:00	15:00 to 23:00	23:00 to 07:00	07:00 to 15:00	15:00 to 23:00	23:00 to 07:00	07:00 to 15:00	15:00 to 23:00	23:00 to 07:00	
Three TP &M Twinpacs	8.816	8.571	4.733	12.71	10.59	10.87	7.277	6.997	3.931	74.45
Five W-251's	9.392	8.869	5.393	12.85	11.48	12.22	8.025	7.813	4.824	80.87
Two W-501's	8.094	7.657	4.894	11.60	10.77	10.58	6.615	6.314	4.263	70.79
One W-501 and Two W-251's	9.050	7.657	4.894	12.56	10.62	9.908	6.615	6.314	4.263	71.88

fuel consumption occurs during the winter day, that the least uniform consumption rate occurs during the summer day, and that the minimum consumption rate occurs during the spring-fall day. It is also evident that the average and extreme fuel requirements for each of these systems is approximately independent of the system considered.

CHAPTER 5

WASTE HEAT RECOVERY UNIT

To recover the waste heat from the gas turbine exhaust an air-to-water heat exchanger is needed. Usually, this type of heat exchanger would have a counterflow-shell and tube configuration. However, because of the sensitivity of gas turbine performance on exhaust back pressure, a high-performance finned-tube crossflow exchanger is necessitated. An estimate of the effect on turbine performance of exhaust back pressure is given in a memorandum[13] from TPM - "A one-inch of water pressure increase in exit loss decreases power about 1/4% and increases the corresponding heat rate 1/8%." This quote applied to TPM's FT4C turbine and is only an approximate estimate. A more accurate presentation of the performance degradation with back pressure is given in Fig. 5.1 where heat rate versus load curves for the FT4C are plotted with 1" H_2O exit loss (normal) and 10" H_2O exit loss (with TPM's waste heat boiler).

In this chapter, a design for a waste heat recovery unit is presented. This exchanger has been designed using performance criteria provided by Waste Heat Engineering Corporation[14] (one of the companies which build TP&M's waste heat boilers for their combined cycle plant). It is important to emphasize that this is not an optimized design (WHECO can provide such a design only once the plant specifications are completely specified); but rather it is a workable design provided to give some idea of the waste heat

recovery units' geometry. The internal arrangement would be similar to that shown in Fig. 5.2 (TPM's waste heat boiler) with the U-tubes interconnected so that the water would enter at the top and exit at the bottom while the exhaust gases enter at the bottom and exit at the top of the heat exchanger.

The following discussion presents a workable design to recover the waste heat generated by one FT4C gas turbine. The decision as to whether each turbine should have its' own heat exchanger or whether one heat exchanger should service multiple turbines would have to be made after the gas turbine system was selected. The required heat exchanger input parameters are the following:

exhaust gas flow = 1,040,400 lb/hr.,
exhaust gas temperature = 829 °F
water flowrate = 395,594 lb/hr.,
water inlet temp. = 40 °F,
water outlet temp. = 385 °F, and
heat transferred = 40 MW.

The design parameters are the following:

Tube diameter = 2.375" (2in. sch. 40 pipe),
pitch = 6 in.,
tube length = 20 feet,
no. of tubes per row = 24,
no. of rows = 17,
fin height = 1 in.,
fin thickness = .03", and
no. of fins per inch = 8.

The thermodynamic parameters are the following:

heat transfer coefficient inside tube = 1378 BTU/hrft^2 °F,

heat transfer coefficient outside tube = 9.56, BTU/hrft^2 °F,

overall heat transfer coefficient (based on total outside area) = 5.94, BTU/hrft^2 °F,

total outside area = 93,922 ft^2,

mean logarithmic temperature difference = 369 °F, and

total heat transferred = 145,078,240 BTU/hr.

This gives a "safety factor" of 1.063 regarding the heat transfer capability of the heat exchanger. Also, it is necessary to note that the total area referenced above includes the fin area and that the total area of the pipe is only 5,077 ft^2. The pressure drop for this heat exchanger is less than 6 in. of water, whereas a shell and tube type would have a pressure drop of about 35 in. of water (approx. 1.25 PSI). Available estimates for the cost of such a heat exchanger are approximately $800,000.

The trade-offs between use of a single large heat exchanger and use of several smaller units are the following:

1) A single heat exchanger is less expensive than several small units,

2) Several exchangers allow the ability to continue to operate at part load if one unit requires maintenance,

3) A single large exchanger takes up less groundspace,

4) Several exchangers allow for more flexibility in the arrangement of the plant, and

5) Several exchangers allow the ability to make adjustments to one unit without affecting the rest of the system, (for example, one exchanger could be utilized to generate steam at low pressures, or could supply water at a variety of temperatures to various users).

It can be argued that the flexibility and added system reliability of multiple heat exchangers overcomes the capital cost differential (which in any event should not be very large) of a single large heat exchanger.

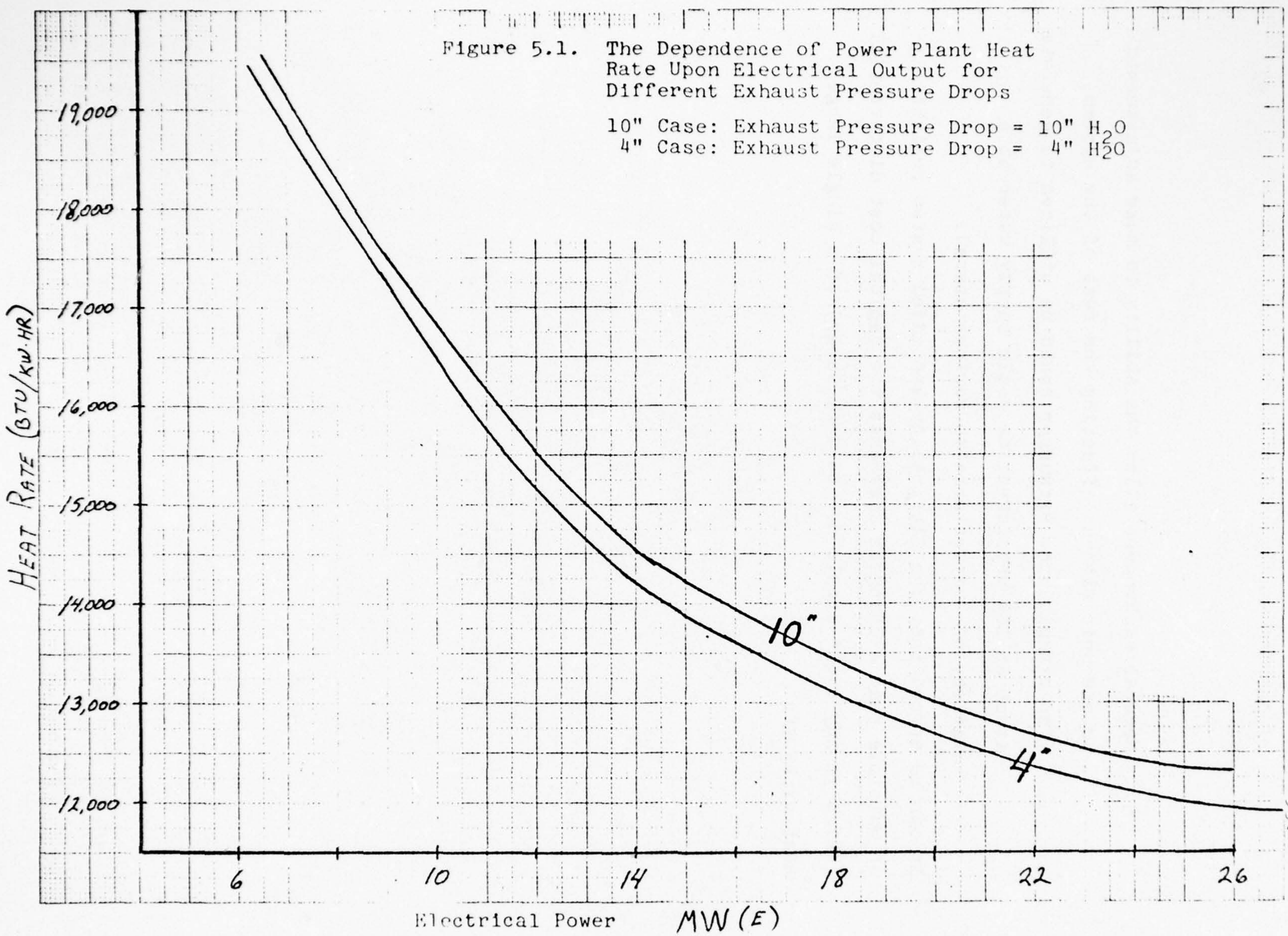


Figure 5.1. The Dependence of Power Plant Heat Rate Upon Electrical Output for Different Exhaust Pressure Drops

10" Case: Exhaust Pressure Drop = 10" H_2O
4" Case: Exhaust Pressure Drop = 4" H_2O

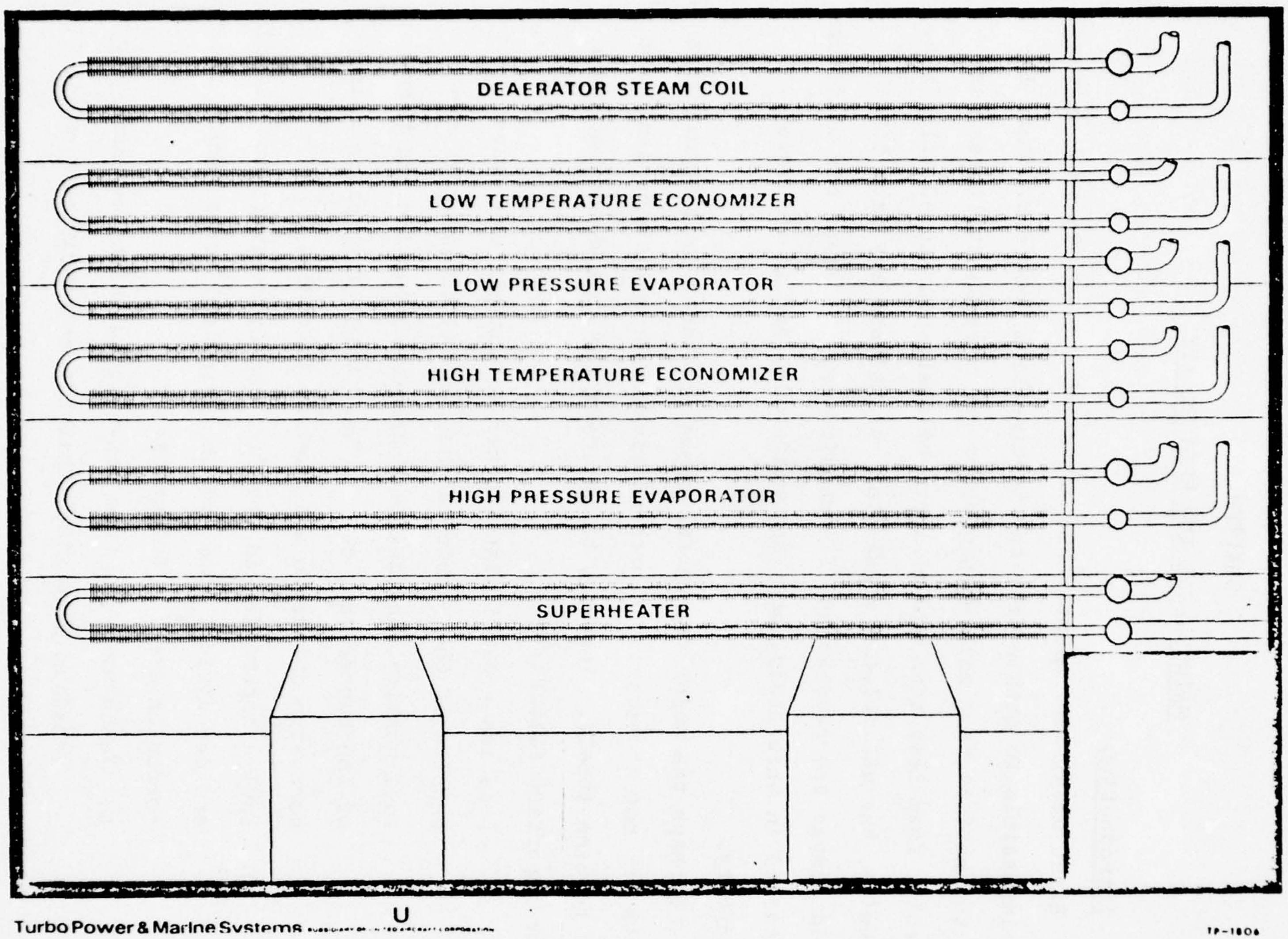


Figure 5.2. Waste Heat Recovery Boiler Designed for Use with TP&M FT4C Gas Turbine

CHAPTER 6

MAINTENANCE AND RELIABILITY

6.1 Introduction

Exact details of maintenance costs and forced-outage down-time are impossible to obtain from the turbine manufacturers. This is due to the fact that maintenance costs for various utilities vary greatly, from less than one mill per kilowatt-hour to 50 mills/kwh. Likewise, the utilities are reluctant to release any data that would damage the reputation of a manufacturer. Out of six utilities contacted in this study, only one responded, and it requested anonymity.

Perhaps the major reason for these attitudes is that maintenance costs are *not necessarily representative of or even dependent upon* the turbine itself. Instead, these costs are more dependent upon four important factors:[15]

1) Location - particulate matter in the air can cause erosion of the turbine combustor and clogging of the fuel nozzles, leading to improper combustion and burn-out of the burner can, also, a salt water atmosphere causes corrosive damage to all turbine internals.

2) Fuel - impurities in the fuel can cause corrosion, clogging, and deposition. Some of the principal contaminants causing corrosion are the following:

   a) Vanadium - can form low-melting point compounds (eg. vanadium pentoxide) that are extremely corrosive.

Hence, vanadium concentrations should be less than 0.5 PPM (wt) and the weight ratio of magnesium (an inhibitor) to vanadium should be maintained at 3.0,

b) Sodium and Potassium - these materials combine with vanadium to increase corrosion. The total amount of sodium and potassium should be less than 0.5 PPM (wt.),

c) Lead - causes corrosion itself and destorys the beneficial effect of magnesium on vanadium. Its presence is usually the result of contamination during processing or transportation and its level should be less than 2 PPM, and

d) Sulfur - causes no corrosion itself but can combine with sodium and potassium to form corrosive compounds. It is impractical to prevent corrosion by controlling the sulfur level, so this type of damage is reduced by limiting the levels of sodium and potassium.

Clogging can be caused by noncombustible material such as solid particles or oil-and/or water-soluble metallic compounds ,

Deposition is the result of materials such as calcium whose deposits are almost impossible to remove, and calcium must be limited to less than 10 PPM,

3) Preventive Maintenance - a well thought-out preventive maintenance schedule and conscientious system monitoring can reduce costs considerably. An example where the merit of this procedure is indicated, is the "burnout"

of burner cans. These relatively inexpensive components need to be replaced every 500-2000 hours; however, an undetected "burn out" can cause extensive damage to the turbine blades through improper combustion. Replacing a row of turbine blades is an expensive undertaking. A calculational method for scheduling maintenance is given in Appendix B, and Appendix C and is TPM's recommended inspection schedule.

4) Turbine Operating Mode - typical procedure for a manufacturer is to equip their systems with three operating modes, Base, Peak and Reserve. The maximum outputs for these three modes are given in Table 6.1 for the three turbines considered in this study. Operation at the peak-mode causes approximately twice the rate of damage as base-mode operation, and the reserve-mode running causes 9 to 12 times the base-mode damage rate. Most utilities never use the reserve mode, and they use the peak mode as seldom as possible (it is sometimes regarded as more economical to run two turbines at half of base load each rather than one turbine at peak load). Because of this, in this study, only use of the base-load mode is considered in any calculations, and it is recommended that use of gas turbines in any other mode be prohibited. Consequently operation and maintenance costs can not be compared solely on a dollar per hour or kw-hour basis (which is how the

the data are reported). Rather a comparison in dollars per hour or kw hour for each operational mode should be performed.

6.2 Maintenance Cost Data

Despite the reluctance of utilities to furnish any data to this study, they are required to file yearly reports of such data with the Federal Power Commission. From these reports,[16] the journal "Gas Turbine International compiles a listing of turbines and their operating records. The data from the 1974 Annual Report are given in Appendix D, but more importantly, these data are illustrated in the probability density curves of maintenance for three manufacturers (GE, Westinghouse, and TP&M) in Figs. 6.1 through 6.6. Two curves are presented for each manufacturer, the first covers the total range of costs, and the second, details the probability density curve up to the average value. The primary fuels used in these turbines have been liquid forms (oil, aviation kerosene, etc.) and it is generally recognized that burning natural gas reduces maintenance costs by a factor in the range from two to four. Also, many of these turbines were not under a good preventive maintenance plan, so the maintenance cost values in these data should be much larger than what could be obtained in an application with a conscientious maintenance program. Figure 6.7 is a probability density curve of the maintenance costs for industrial type gas turbines running on natural gas (data are from Sawyer's Gas Turbine Handbook, Vol. 3, 1972).[17]

The data from the one utility that complied with the request of this study are plotted on Figure 6.1. All of their turbines are TP&M twin paks run on aviation kerosene. Their costs are plotted as two points on the probability curve for TP&M, first, the costs before a comprehensive maintenance plan was instituted (1970-74) and second, after institution of such a program (1975).

Trying to summarize the probability data in a single average cost value, in order to facilitate comparison among turbines is difficult; however, if the formula

$$(\text{Avg. Cost} \cdot \frac{\text{mills}}{\text{kwhr}}) = \Sigma(\frac{\text{cost}}{\text{hr}}) \cdot \frac{\Sigma(\text{hours run})}{\Sigma(\text{kilowatt-hrs})}$$

is used the resulting average costs are the following:

G.E. - 1.84 mills/kw-hr

TP&M - 2.12 mills/kw-hr

Westinghouse - 3.08 mills/kw-hr

6.3 System Reliability

In the utility experience survey in this study it was impossible to obtain any direct information regarding reliability, other than testimony to the effect that it's about 95-97%.[17] However, given that all turbines are likely to fall within a few percentage points of each other in individual reliability, another factor becomes more important and that is the maximum capability of a system when one or more turbines is down. For example, in considering the design of a 100 MWe system, and comparing a W-501

and two TP&M twin pacs, one would discover that about 5% of the time the W-501 could produce zero electricity and that the TP&M twinpaks could produce 75 MWe. In an application where redundancy is necessitated, a system that has partial redundancy built-in would clearly be preferable. Table 6.2 compared the four 140 MWe systems examined in Chapter 4 in terms of the maximum electric output possible with one or more turbines off-line.

6.4 Utility Recommendations

Although the utility personnel interviewed were unwilling to provide any operational data, almost all were willing to give their opinions (anonymously) regarding which system (out of G.E., Westinghouse and TP&M) would be best for the application of interest in this study. Of the six utility representatives contacted, one refused to comment, one recommended G.E. units, and four recommended TP&M units.[17]

6.5 Ease of Repair

For minor repairs, all the turbines are basically equally easy to repair, since this requirements in considered to be important in their design. For major repairs, however, a large difference exists. Industrial type turbines are generally repaired on site, whereas, aircraft derivative turbines are repaired at a maintenance center. Due to its small size and modular construction, either the gas generator or the free turbine of a TP&M turbine can be shipped to a center where it can be completely

overhauled in a period which typically varies between one and four weeks. While data regarding overhaul of a Westinghouse turbine was difficult to obtain, one utility representative observed that in at least one case required major repairs took several months. All of the utility representatives agreed that repair of the TP&M turbines was easier and quicker than with the industrial type turbines.

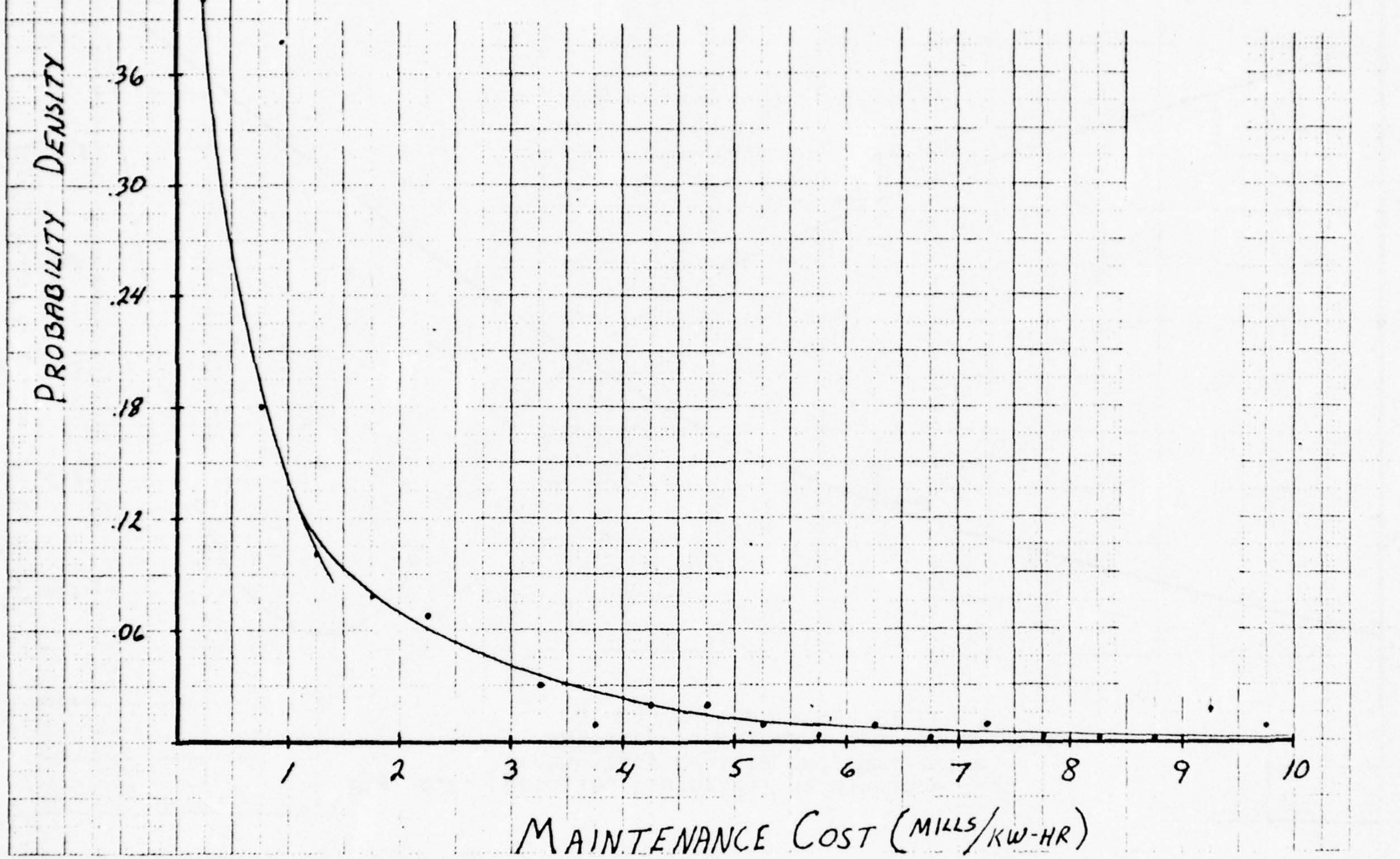


Figure 6.1. Historical Maintenance Cost Probability Density Function - General Electric Co. Gas Turbine Power Plants

Fig. 6.2. Detailed Historical Maintenance Cost Probability Density and Cumulative Probability Functions - General Electric Co. Gas Turbine Power Plants

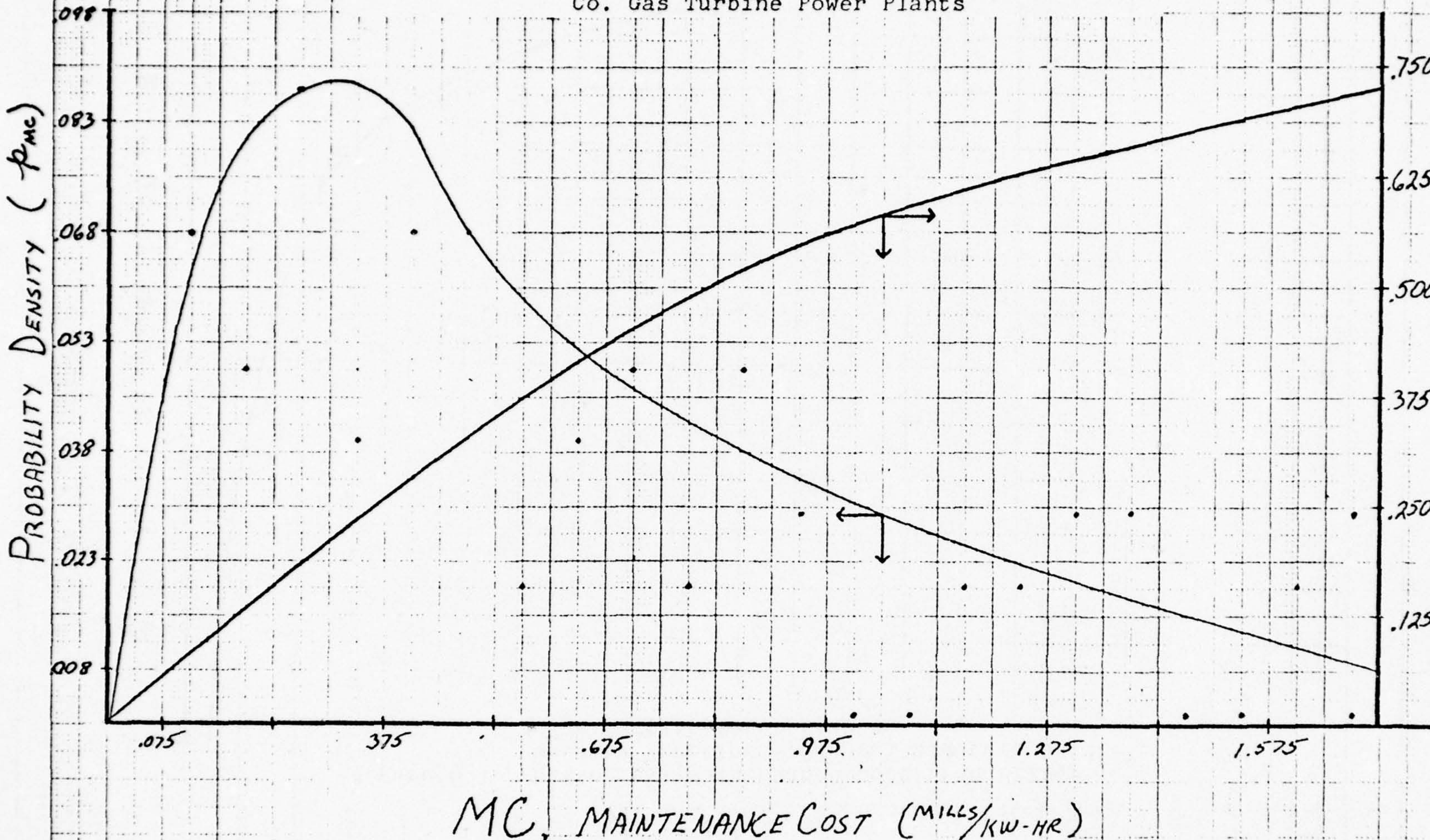


Figure 6.3. Historical Maintenance Cost Probability Density Function - Westinghouse Electric Corp. Gas Turbines

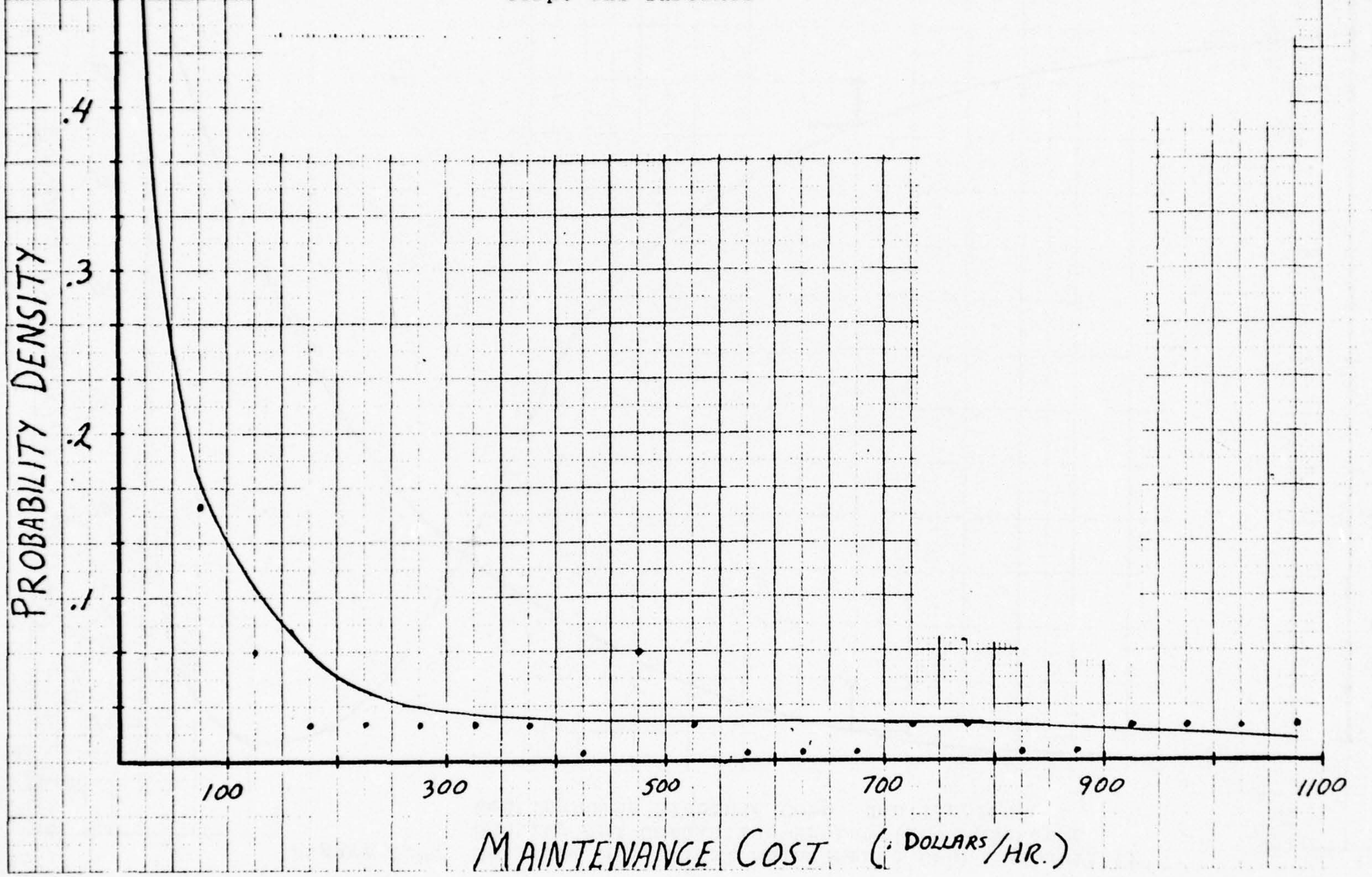


Figure 6.4. Detailed Historical Maintenance Cost Probability Density and Cumulative Probability Functions - Westinghouse Electric Corp. Gas Turbines

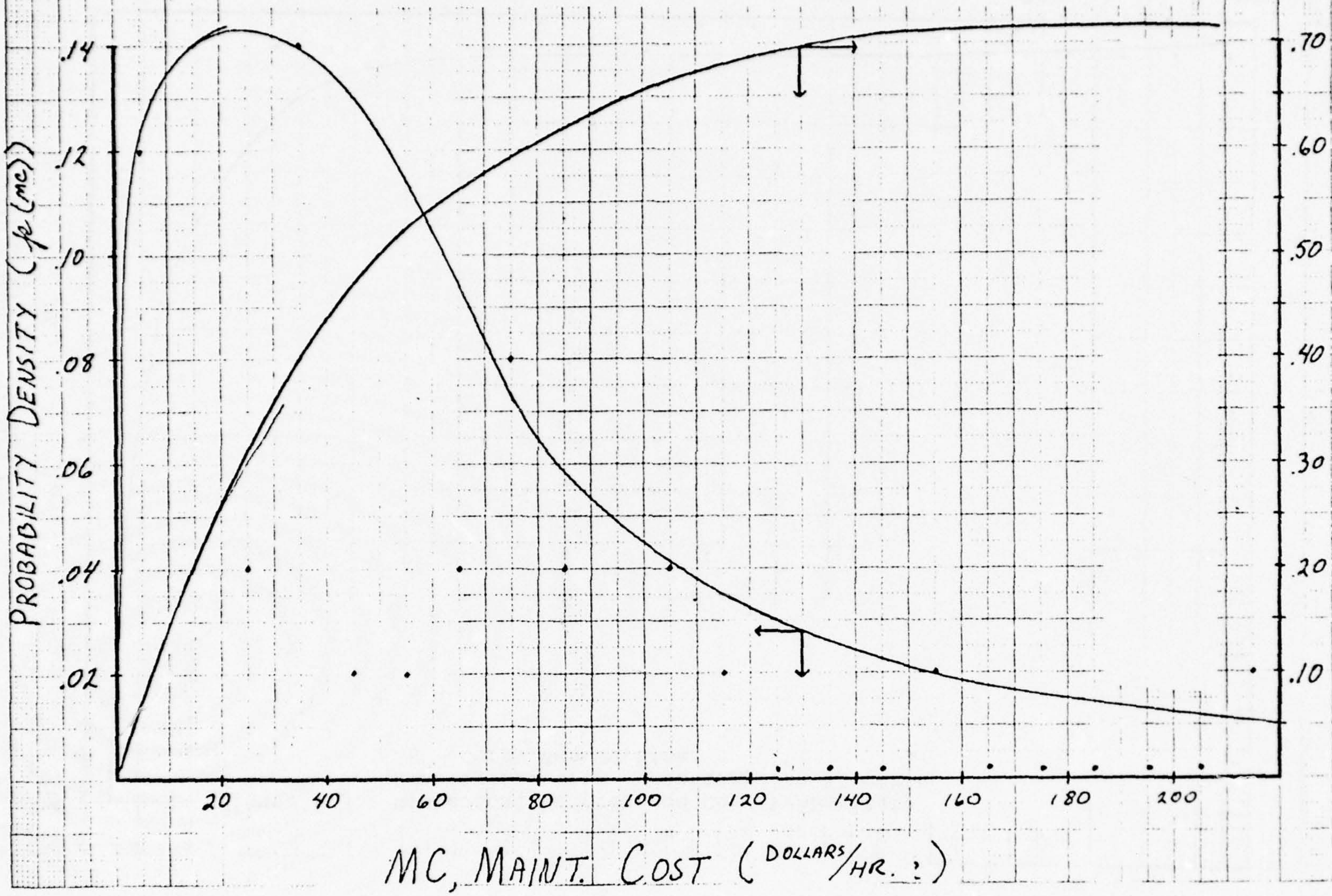


Figure 6.5. Historical Maintenance Cost Probability Density Function - Turbo Power and Marine Gas Turbines

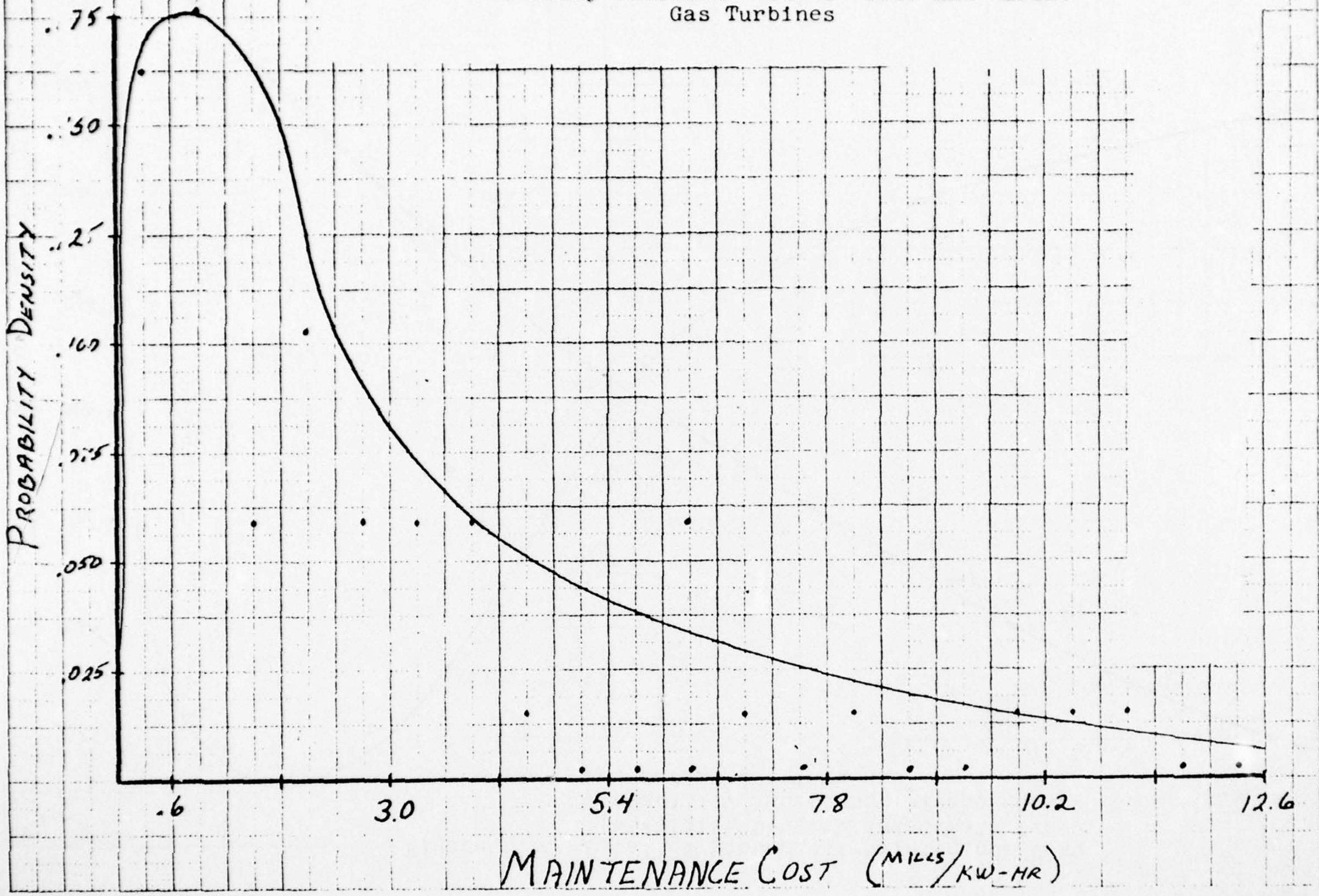
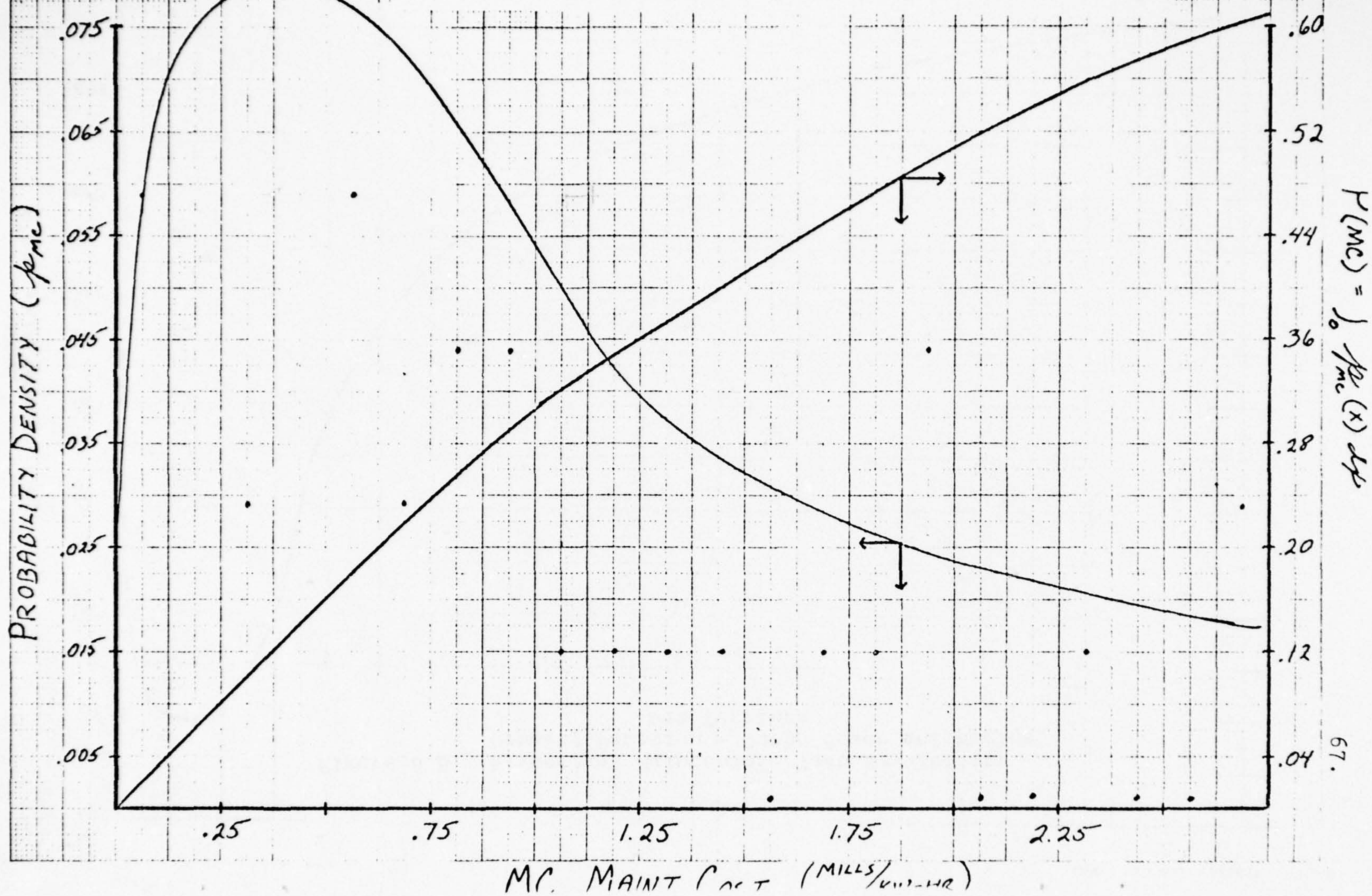


Figure 6.6. Detailed Historical Maintenance Cost Probability Density and Cumulative Probability Functions - Turbo Power and Marine Gas Turbines



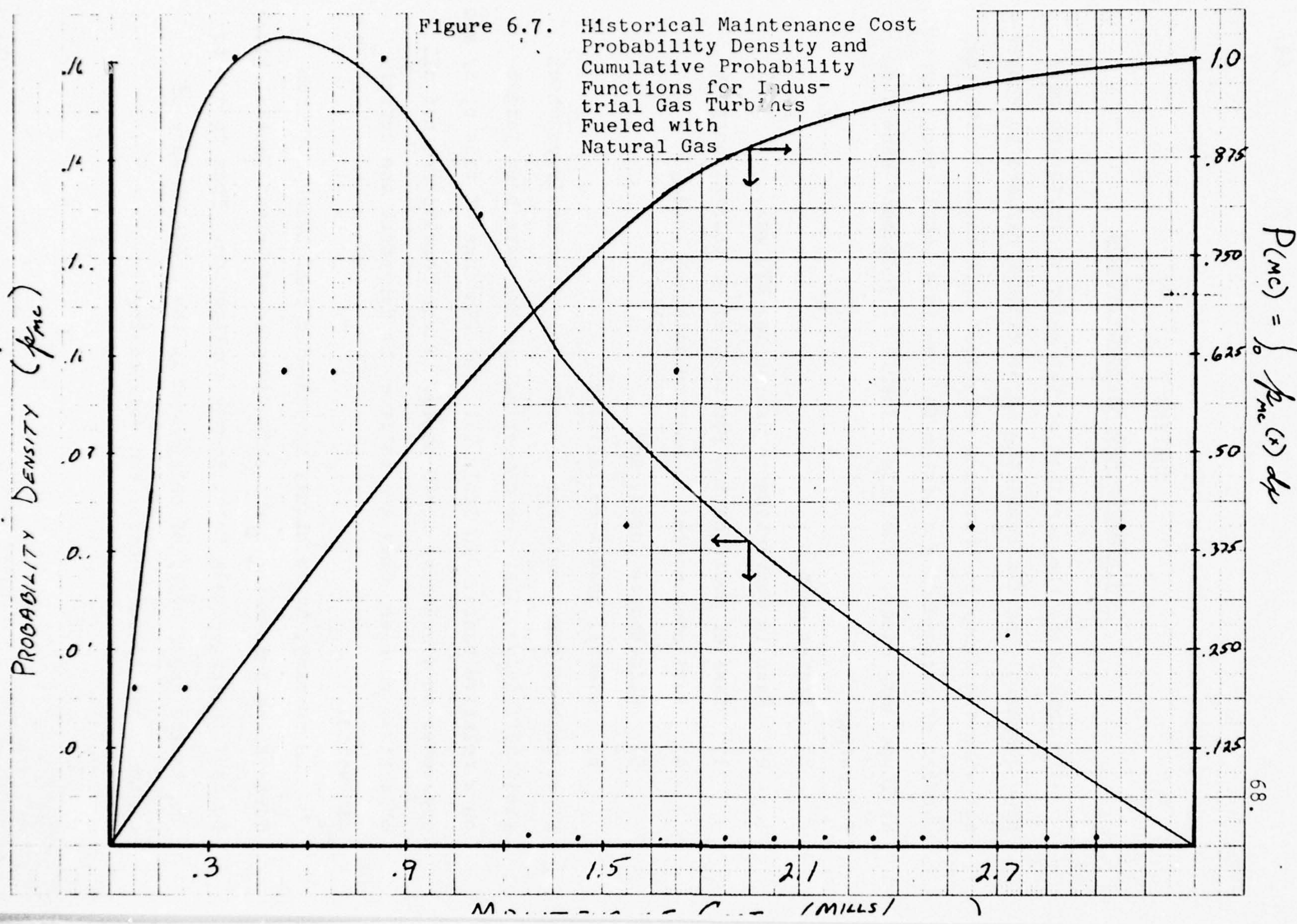


Figure 6.7. Historical Maintenance Cost Probability Density and Cumulative Probability Functions for Industrial Gas Turbines Fueled with Natural Gas

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

In this study the turbines of two major manfacturers have been compared (the two remaining major vendors refused to cooperate with the study in providing requested information) with the purpose of recommending a system for use in a total energy system at large Army bases. The main criteria of judgement were the following:

1) System cost,
2) Ease in converting to use of low BTU gas,
3) Ease of modification for control of thermal output,
4) Fuel consumption in meeting seasonal loads,
5) Maintenance costs, and
6) Overall system reliability.

Since most of these criteria are not subject to analyses which yield definitive data, the competing systems have been judged on a relative basis. In Table 7.1, the findings of this study are presented on a relative merit basis. These results are not all objective; however, they are believed to represent the actual situation.

In summary, it is highly recommended that twin-pacs from Turbo Power and Marine Systems (a division of United Technologies) be used in large scale Total Energy application. This is due to the marked superiority of this system in the areas of ease of control, maintenance costs, and reliability.

TABLE 7.1. RELATIVE MERITS OF THE COMPETING TURBINES EXAMINED IN THIS STUDY

Criterion / System	System Cost	Low BTU Gas Progress	Thermal Control	Fuel Usage	Maintenance Cost	Overall Reliability
3 TP&M Twin Paks	-1	0	+2	0	+2	+2
2 W-501's	+1	+1	0	+2	-1	0
5 W-251's	0	+1	0	-1	-1	+2
1 W-501 and 2 W-251's	+2	+1	0	+1	-1	+1

Key: +2 - Superior
+1 - Good
0 - Fair
-1 - Poor

TABLE 7.2. SYSTEM RESERVE CAPABILITY FOR A 140 MWe STATION AS A FUNCTION OF THE NUMBER OF INOPERABLE TURBINES (IN UNITS OF MWe)

# of turbines off-line (MW) / Systems	One	Two	Three	Four	Five
3 TP&M Twin Paks	119.7	93.4	73	46.7	26.3
2 W-501's	97.8	0	0	0	0
5 W-251's	134.5	100.8	67.2	33.6	0
1 W-501 and 2 W-251's	67.2 or 131.4	33.6 or 97.8	0	0	0

APPENDIX A

OFF PEAK CALCULATION METHOD

Reference: "A Rapid Calculation Method for the 'Off-Design' Performance of Compressors and Turbines," H.J. Horlock, Gas Turbine Laboratory Report No. 40, 1957.[7]

For a compressor or turbine stage in which the outlet angles relative to stator and rotor remain constant independent of the entry flow coefficient, the increase in stagnation enthalpy is given as

$$\begin{aligned}\Delta h_o &= Cp\Delta T_o = U\Delta C_u, \\ &= U(C_{x_2}\tan\alpha_2 - C_{x_1}\tan\alpha_1), \qquad \text{(A.1)} \\ &\doteqdot U(U - C_{x_1}(\tan\alpha_1 - \tan\beta_2)), \\ &\text{if } C_{x_1} \doteqdot C_{x_2};\end{aligned}$$

and

$$\psi = \frac{Cp\Delta T_o}{U^2} = 1 - \phi t \quad ,$$

where $\phi = C_x/U$ - the flow coefficient, and $t = \tan\alpha_1 - \tan\beta_1$ = a constant.

Assuming little change in kinetic energy from entry to exit, then it is seen that $\Delta T_o \doteqdot \Delta T$, and

$$\psi = \frac{Cp\Delta T_o}{U^2} = \frac{Cp\Delta T}{U^2} \quad . \qquad \text{(A.2)}$$

Let the design point be designated (ψ^*, ϕ^*), where

$$\psi^* = 1 - \phi^* t \tag{A.3}$$

Then at off-design operating points

$\psi = 1 - \phi t$, and the result

$$\frac{\psi}{\psi^*} = \frac{1}{\psi^*} - \left(\frac{1-\psi^*}{\psi^*}\right) \frac{\phi}{\phi^*} \tag{A.4}$$

is obtained.

Now we idealize this relation to a stage for which the temperature rise is infinitesimally small, but the nature of the off-design performance must not be changed, i.e., ψ^* must remain unchanged. The actual machine may be replaced by a machine which has an infinite number of small stages of infinitesimally small axial length; the geometry of the blading of one of these small stages is the same as that of the real stage at the same axial location (i.e. t and ψ^* are the same). Let m be the number of small stages, and $\Delta T'$ be the temperature rise, then

$$m\Delta T' = \Delta T \tag{A.5}$$

and

$$\psi = \frac{\Delta T}{U^2} = \frac{\Delta T'}{(U')^2}, \quad U' = \frac{U}{\sqrt{m}} . \tag{A.6}$$

Since ϕ is unchanged, one obtains the result

$$\frac{C_x}{U} = \frac{C'_x}{U'}\ , \quad \text{where } C'_x = \frac{C_x}{\sqrt{m}}\ . \tag{A.7}$$

For the same mass flowrate to be passed $\sqrt{m}$ machines must be used in paralle, and Eq. A.4 can be rewritten in the form

$$\left(\frac{U^*}{U}\right)^2 \frac{dT}{dT^*} = \frac{1}{\psi^*} - \left(\frac{1-\psi^*}{\psi^*}\right)\frac{\phi}{\phi^*}\ , \tag{A.8}$$

where T^*, ψ^*, ϕ^*, U^* are distributions through the machine at the design point, and T, ϕ, U are the distributions through the machine in the "off-design" state.

At the design point

$$C_x^* A^* \rho^* = C_{x_I}^* A_I^* \rho_I^*\ , \tag{A.9}$$

where I represents conditions at the entrance to the machine. For a machine operating off-design, but with the same entry conditions ($\rho_I = \rho_I^*$, $T_I = T_I^*$) ,

$$\begin{aligned} C_x A\rho &= C_{x_I}\ \rho_I A_I \\ &= C_{x_I} \rho_I^* A_I^*\ . \end{aligned} \tag{A.10}$$

From Eqs. (A.9) and (A.10) it is seen that

$$\frac{C_x^* \rho^*}{C_x \rho} = \frac{C_{x_I}^*}{C_{x_I}} = \frac{M^*}{M} \quad (M = \text{mass flow rate})\ . \tag{A.11}$$

If η_p is the polytropic efficiency, then it is seen that

$$\left(\frac{T^*}{T_I^*}\right)^n = \left(\frac{\rho^*}{\rho_I^*}\right) \quad , \tag{A.12}$$

where $n = \left(\frac{k}{k-1}\right) \eta_p^{\pm 1} - 1$. (+1 for compressors, -1 for turbines).

Consequently the result

$$\left(\frac{T}{T_I}\right)^n = \left(\frac{T}{T_I^*}\right)^n = \frac{\rho}{\rho_I} = \frac{\rho}{\rho_I^*} \tag{A.13}$$

is obtained.

From Eqs. (A.12) and (A.13) it is seen that

$$\left(\frac{T}{T^*}\right)^n = \frac{\rho}{\rho^*} \quad . \tag{A.14}$$

From Eqs. (A.11) and BA.14), the result

$$\frac{C_x}{C_x^*} = \left(\frac{M}{M^*}\right)\left(\frac{\rho^*}{\rho}\right) = \xi\left(\frac{T^*}{T}\right)^n \tag{A.15}$$

$(\xi = \frac{M}{M^*})$ is obtained.

From Eqs. (A.8) and (A.15) it is seen that

$$\left(\frac{U^*}{U}\right)^2 \frac{dT}{dT^*} = \frac{1}{\psi^*} - \left(\frac{1-\psi^*}{\psi^*}\right) \frac{\xi U^*}{U} \left(\frac{T^*}{T}\right)^n \quad . \tag{A.16}$$

Writing $\lambda = U^*/U$, the result

$$\frac{dT}{dT^*} = \frac{1}{\psi^* \lambda^2} - \left(\frac{1-\psi^*}{\psi^*}\right) \frac{\xi}{\lambda} \left(\frac{T^*}{T}\right)^n \tag{A.17}$$

is obtained.

Thus, the temperature distribution through the machine (T, the dependent variable) is expressed in terms of the temperature distribution through the machine at the design point (T*, the independent variable).

Writing $u = T/T^*$ and differentiating, one obtains the expression

$$\frac{dT}{dT^*} = U + T^* \frac{du}{dT^*} \tag{A.18}$$

From Eqs. (A.17) and (A.18) is is seen that

$$u + T^* \frac{du}{dT^*} = A + \frac{B}{u^n} \quad , \tag{A.19}$$

where

$$A = \frac{1}{\psi^* \lambda^2} \quad \text{and}$$

$$B = \left(\frac{\psi^* - 1}{\psi}\right) \frac{\xi}{\lambda} \quad .$$

The solution of the differential equation (A.19) can be found assuming ψ^* constant and approximating u by $(1+\varepsilon)$ where ε is small. Then Eq. (A.19) transforms to

$$\frac{dT^*}{T^*} = \frac{u^n du}{Au^n + B - u^{n+1}}$$

$$= \frac{(1+n\varepsilon)d\varepsilon}{A(1+n\varepsilon)+B-1-(n+1)\varepsilon} \quad (A.20)$$

Integrating this expression (with $\varepsilon_I = 0$) one obtains the result

$$\ln(\frac{T_{II}}{T_I}) = \frac{n\varepsilon_{II}}{nA-(n+1)} - \frac{nB+1}{(nA-(n+1))^2} \cdot$$

$$\log_e[1 + (\frac{nA-(n+1)}{A+B-1})\varepsilon_{II}] \quad (A.21)$$

Thus, if the design temperature ratio is known, Eq. (A.14) may be solved for the off-design temperature ratio as

$$\frac{T_{II}}{T_I} = \frac{T_{II}}{T^*_{II}} \cdot \frac{T^*_{II}}{T^*_I} = \frac{T^*_{II}}{T^*_I}(1 + \varepsilon_{II}) \quad . \quad (A.22)$$

This is the method used to predict the off-design performance in this study.

Sample Problem

TP&M turbine using bypass control scheme, operating at full power level.

$$\left.\frac{M}{M^*}\right)_{\text{gas generator}} = 1 \quad : \quad \left.\frac{M}{M^*}\right)_{\text{free turb.}} = .75$$

For the free turbine the following data obtain:

$\eta_p = .846$ $\qquad$ $\psi^* = -1.953$

$k = 1.370$ $\qquad$ $\ln(T_{II}/T_I)^* = -.2249$

$(\Delta h)^* = 88.92$ $\qquad$ $\lambda = U^*/U = 1$

$M^* = 1040400$ lbm/hr. $\qquad$ $\xi = M/M^* = .75$

$U^* = 1068.14$ ft/sec ,

and:

$$A = 1/(\psi^*\lambda^2) = -.5120$$

$$B = \left(\frac{\psi^*-1}{\psi^*}\right)\frac{\xi}{\lambda} = (1.512)\cdot\xi = 1.134$$

$$n = \left(\frac{k}{(k-1)}\right)\left(\frac{1}{\eta_p}\right) - 1 = 3.377 \quad .$$

The equation that predicts system performance is:

$$\ln\left(\frac{T_{II}}{T_I}\right)^* = \frac{n\varepsilon_2}{nA-(n+1)} - \frac{nB+1}{(nA-(n+1))^2}\cdot\ln\left(1+\left(\frac{nA-(n+1)}{A+B-1}\right)\varepsilon_2\right)$$

Substituting in the above values one obtains the expression

$$-.2249 = (-.5531)\varepsilon_2 - (.1295)\ln(1+(16.15)\varepsilon_2) \quad .$$

This equation cannot be solved directly for ε_2, but an iterative procedure gives the results $\varepsilon_2 = .135$, and

$$\frac{T_{II}}{T_I} = (1+\varepsilon_2)\cdot\left(\frac{T_{II}}{T_I}\right)^{*}, \text{ where } T_I^* = 1745\ °F,$$
$$T_{II}^* = 829\ °F\ .$$

The value of T_I remains equal to that of T_I^*, since the turbine is running at full power. Therefore it is seen that

$$T_{II} = (1.135)\cdot(829+460) - 460$$
$$= 1003\ °F, \text{ and}$$
$$\Delta h = 41.75 \text{ BTU/lbm}\ .$$

The electric power produced is given by (with generator efficiency = .97) the expression

$$\dot{E} = (.97)(41.75)(.75)(1040400)/(3412)$$
$$= 9{,}262 \text{ KW}$$

The waste heat production rate is given by the sum of the enthalpy fluxes of the bypass flow and the turbine exhaust (referenced to a heat exchanger exit temperature of 348 °F) as

$$\dot{Q}_{WASTE} = \frac{(1040400)}{(3412)}((.25)(412.6)+(.75)(370.8)-203.4)$$
$$= 54{,}231.\ \text{KW}\ .$$

TABLE A.1

LIST OF SYMBOLS

h_o	= stagnation enthalpy	
c_p	= specific heat at constant pressure	
T_o	= stagnation temperature	
T	= static temperature	
ρ	= density	
p	= pressure	
α_1, α_2	= air angles (absolute)	
β_1, β_2	= air angles (relative)	
t	= $\tan\alpha_1 - \tan\beta_2$	
c_x	= axial velocity	
c_u	= tangential velocity	
U	= blade speed	
$\psi = \frac{Cp\Delta T_o}{U^2}$	= stage loading	
$\phi = C_x/U$	= flow coefficient	
A	= area	
M	= mass flow rate	
ξ	= M/M^*	constants
λ	= U^*/U	constants
A, B, b		constants
η_p	= polytropic, or small stage, efficiency	
k	= ratio of specific heats	
n	= $\left(\frac{k}{k-1}\right)\eta_p^{\pm 1} - 1$	
u	= $T/T^* = 1 + \varepsilon$	
r	= pressure ratio	

TABLE A.1 (continued)

F $= \dfrac{\psi/\psi^*}{\phi/\phi^*}$

Subscripts

I entry to machines

II exit from machines

Superscripts

$*$ design point, condition of maximum efficiency

APPENDIX B

WESTINGHOUSE MAINTENANCE PROCEDURES

The enclosed material was provided by the Westinghouse Electric Co. for use in this study, and it discusses maintenance procedures and recommendations for Westinghouse gas turbine units.

RELIABILITY

MAINTENANCE AND MAINTENANCE COSTS

Maintenance and its cost are objects of justifiable concern to gas turbine operators. Specifically, operators are concerned with two aspects of the maintenance cost question—accurate prediction of the cost over a period of time and keeping the overall cost as low as possible.

Since maintenance is such a critical item to operators, it automatically becomes a primary concern of manufacturers. For this reason, a great deal of effort has been directed towards attempting to establish realistic maintenance schedules and keeping costs—commonly measured in dollars per fired hour—to a minimum. The two items are, of course, closely associated; with maintenance costs being a function of maintenance intervals. Thus, keeping costs down is largely a matter of determining the longest practical maintenance interval that is commensurate with safe and efficient operation.

Maintenance intervals are most logically determined as a function of part life since it sets the interval between parts replacement requirements. A value for this part life can be arrived at basically in one of two ways:

- Continue to operate until a part fails, replace the part and run until the next failure.
- Determine scheduled intervals at which parts will automatically be replaced in a program of preventive maintenance.

While the first plan may result in the fewest total replacements over a period of time, it may also prove to be a costly practice. The unscheduled shutdowns and forced outages that are inevitable with such a plan will seldom occur at the operator's convenience. Quite the opposite is usually the case, making a "crash" program in repair, and the need to purchase outside power for a time, a real possibility. Such situations can easily wipe out the savings resulting from the extended intervals.

If the attempt is made to circumvent such problems by establishing a replacement schedule, the question is immediately raised as to what criteria should be used to estimate parts life. This a complex problem which is extremely difficult to answer analytically. However, certain criteria have been established and Westinghouse has built predictions around them in an effort to provide its customers with quantitative data on maintenance requirements.

Calculating Part Life

For gas turbines that are operated under essentially continuous service conditions, parts life is considered to be a function of metal "creep", which in turn is directly related to engine firing level. The firing level in a gas turbine can be broadly considered as falling into one of three categories—base load, peak load, and

system reserve. These levels are constantly increasing as turbine technology advances, but in the latest Westinghouse gas turbines, they correspond to temperatures of 1870°, 1950°, and 2010°F respectively.

This firing level categorization is observed in the creep life fraction equation which is used by Westinghouse to measure part life. The equation takes the following form:

$$\frac{\text{Base Fired Hours}}{X} + \frac{\text{Peak Fired Hours}}{Y} + \frac{\text{System Reserve Fired Hours}}{Z}$$

with the creep life constants X, Y, and Z varying for each part and for the type of fuel (gas or oil) used. The equation assumes that when the sum of the life fractions for the individual modes of operation is equal to one, the life of the part has been expended.

Westinghouse engineering has established creep constant values for the various engine hot-end parts–blades, vanes, combustor baskets, transition pieces. These values are based on calculated creep rupture life and field experience. The values are continually being expanded and updated as work in this area progresses and further data becomes available. This process ensures that Westinghouse is in a position to provide its customers with the most reliable information possible in this area at any given time.

The Influence of Cycling

Unfortunately, the value of the creep life fraction equation, by itself, as a guide to gas turbine part life is limited. It assumes continuous duty with few starts and stops, definitely not a typical operating condition for such engines. Gas turbines are most commonly used for intermittent duty such as peak shaving, in which a great deal of cycling is required, and the effect of this cycling must be taken into account. With the existing state of the art, accurately predicting part life under such conditions is more of an empirical than an analytical process, requiring a good deal of field operating experience. However, Westinghouse has developed an analytical method for predicting the life of parts based on a combination of engineering evaluation and experience.

The method establishes a Life Factor and a Cyclic Factor for each part. The Life Factor is based on operating hours at various firing levels; the Cyclic Factor is based on the frequency and rapidity of starts. Both factors are used in conjunction with a Parts Life Limit Chart, shown in Figure 7-1.

The Life Factor is identical to the left-hand side of the creep equation presented earlier, or:

$$LF = \frac{\text{Base Fired Hours}}{X} + \frac{\text{Peak Fired Hours}}{Y} + \frac{\text{System Reserve Fired Hours}}{Z}$$

with the creep life constants remaining the same.

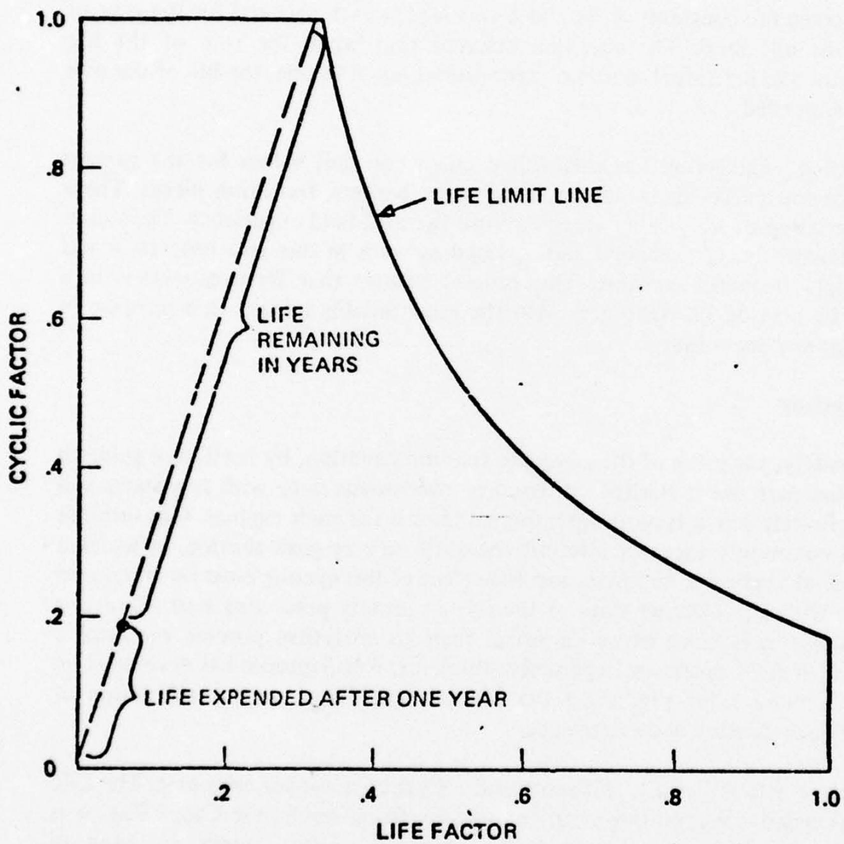


Figure 7-1 Parts Life Limit Chart

The Cyclic Factor equation is similar in form:

$$CF = \frac{\text{No. Normal Starts}}{A} + \frac{\text{No. Fast Starts}}{B} + \frac{\text{No. Emergency Starts}}{C}$$

with the cyclic life constants A, B, and C also varying for each part and with the type of fuel used.

To determine the operating life of a component, the Life and Cyclic Factors are calculated for the required operating conditions and the point that they establish is plotted on the chart. A point on the life limit line signifies that the life of the part is considered to be expended.

Using this method, it is possible to monitor powerplant condition by tracking each critical part on a separate chart, plotting points at different times in its operating life. If the cycling and firing temperature modes remain constant, the time when the part will intersect the line can be extrapolated and replacement can be scheduled accordingly.

Determining Maintenance Intervals

As previously mentioned, maintenance intervals are logically a function of parts life. With an analytical tool such as the Life Limit Chart available, a serious effort can be made to establish safe minimum inspection intervals. Westinghouse has done this, calculating intervals that are based on the chart's creep and cyclic factors.

Creep Influence

On the basis of field experience and the study of creep factor constants, Westinghouse has concluded that, compared to base load operation, running at Peak can be expected to decrease part life by a factor of 2 and System Reserve operation shortens life by a factor of 10. Thus, the cumulative effect of mixed operation can be expressed as:

$$a + 2b + 10c$$

where the constants a, b, and c are the number of hours of operation at Base, Peak, and System Reserve, respectively.

On the basis of the same data, Westinghouse also identified the first-row turbine vanes as the parts with the shortest life expectancy, and thus the parts that should be the limiting influence on minimizing inspection frequency. Their life is assessed at 30,000 hours, so assuming that they will be replaced at a Major Inspection, the time for such an inspection can be set as:

$$a + 2b + 10c = 30,000$$

On the assumption that a safe interval for the turbine section is half the life of the most critical part, the following formula results:

$$a + 2b + 10c = 15{,}000$$

Setting the inspection interval for the combustion section as half that of the turbine section yields:

$$a + 2b + 10c = 7{,}500$$

These formulas are based on natural gas. For oil-burning gas turbines, the figures must be modified to reflect the lower part life expectancy that is typical with such fuel. Consideration of this element by Westinghouse points to a 0.8 multiplier as being appropriate, yielding the following formulas for oil-fired operation:

Major Inspection:	$a + 2b + 10c = 24{,}000$
Turbine Inspection:	$a + 2b + 10c = 12{,}000$
Combustor Inspection:	$a + 2b + 10c = 6000$

The Cyclic Factor

The above maintenance interval formulas are based solely on creep factor considerations, with no allowance for the thermal impact of frequent cycling. Compensation for the influence of cycling in Westinghouse calculations provided by including a cycling factor (F) that varies directly with the number of hours per start. Including this factor modifies the formulas for gas-fired operation as follows:

Major Inspection:	$a + 2b + 10c = 30{,}000.\ (F)$
Turbine Inspection:	$a + 2b + 10c = 15{,}000.\ (F)$
Combustor Inspection:	$a + 2b + 10c = 7500.\ (F)$

Formulas for oil-fired operation are similarly altered.

The cycling factor varies from 1 for the operating range of 50 hours per start and above (indicating, of course, that cycling has no discernible effect on part life at that rate) to a figure of about 0.5 for the condition of 5 hours per start and below.

MAINTENANCE AND SERVICE

The effective operation of a gas turbine, as is the case with most complex mechanical devices, is dependent on a scheduled program of maintenance and service, including periodic inspection, repair, and replacement of parts. The operating environment and conditions to which gas turbine components are subjected make such servicing mandatory if the expense and inconvenience of unscheduled outages are to be avoided.

The parts that establish the maintenance requirements of the engine are the hot-gas path parts. These include the combustor baskets, transition pieces, turbine blades and vanes. Other components—those lying in the so-called "cold end" path—are in a more favorable environment and are not a major maintenance concern and require little or no maintenance.

Inspection

The first step in ensuring reliable operation is to establish a pattern of inspections to reveal any need for corrective action as a form of preventive maintenance. A program that has been found to be effective with gas turbine installations is to schedule four categories of inspection of progressive degrees of thoroughness, with intervals depending on operating time and condition. In order of increasing thoroughness, these inspections are as follows:

- Running Inspection
- Combustion System Inspection
- Turbine System Inspection
- Major Inspection

Running Inspection

Running inspections are the simplest, consisting of general observations made while an engine is in service. With attended units, such observations would be made at least daily, and more likely several times each shift. Unattended units on continuous duty would probably receive a running inspection once a week; those on intermittent applications, once a month, although these intervals represent a minimum program.

Observations during this inspection should include temperatures, pressures, vibration, and the integrity of various service systems. The general relationship between load and exhaust temperature should be noted and recorded for future reference. Deviations in this relationship can be an early indication of deterioration of internal parts, excessive leakage, or compressor fouling. In the same way, deviation in the relationship between fuel flow and load should be considered as a possible trouble indicator. Similarly, increased vibration, whether it occurs rapidly or in a continuous trend, may be an early warning that corrective maintenance is required.

Probably the most important running inspection that can be made is the observation of the temperature monitoring systems of both the exhaust and the turbine inlet. Again, significant changes, in either temperature relationships or absolute values, for given operating conditions may be an indication of developing problems.

Every Westinghouse gas turbine comes with an instruction book that contains all of the necessary information to perform the recommended running checks. Because of their significance in reducing overall maintenance costs, maintaining high reliability, and speeding troubleshooting, these checks should be done as and when required by the maintenance schedule.

Combustion System Inspection

More comprehensive than the running check is the combustion system inspection, which involves a short shutdown of the engine to check the condition of the fuel nozzles, igniters, combustors, and the first stage turbine vanes. These are the components that typically are the first to require servicing, making frequent examination of them important.

Turbine Section Inspection

The third category of inspection is a check of the turbine section, with the entire hot gas path being examined. In addition to a repeat of the combustion system inspection, the turbine cylinder cover is removed and the turbine blades and stator vanes are checked. (The bearing, however, can be inspected without dismantling the engine.)

Major Inspection

The most comprehensive scheduled inspection of the engine is the major inspection, occurring at the longest interval. During this check, both the combustor and turbine inspections are performed and compressor blading and all bearings are examined as well. In addition, a complete alignment check, control system checkout, and major inspection of components as recommended by suppliers should be conducted.

Inspection Frequency Criteria

The frequency with which preventive maintenance inspections should be performed is a function of the conditions under which the powerplant is operated. The factors that have the greatest influence on inspection requirements are the following:

- Thermal cycles
- Type of fuel
- Operating temperature
- Environment
- Maintenance practices

Thermal Cycles

Since it influences part life so dramatically, the most significant factor affecting maintenance interval determination is the thermal cycling to which the turbine is subjected. Because each startup and shutdown of a gas turbine engine subjects the hot-gas parts to a great deal of life-shortening thermal stress, the number of starts has a direct effect on inspection frequency. Westinghouse control systems are designed to minimize the thermal shock resulting from frequent starts, but the fact is that any gas turbine that undergoes this kind of duty will exhibit shorter parts life, and thus require more frequent checks, than will one that is in more continuous operation.

The effect of load cycling that involves frequent and rapid load changes is similar to the effect of frequent starts and shutdowns. It results in high thermal stresses that shorten parts life and thus increase the need for inspection and maintenance.

Fuel Type

The influence of fuel on maintenance requirements is a function of the uniformity of the temperature pattern out of the combustor that is possible with a particular type of fuel and the degree of corrosion introduced by its use. Because it burns cleanly and evenly, and is virtually free of corrosive elements, natural gas allows the longest parts life. Distillate oils provide the next highest life expectancy, followed by crude and residual oils, both of which make achieving a uniform temperature profile difficult as well as generally containing a high percentage of corrosive elements.

Operating Temperature

Another factor that affects maintenance intervals is the temperature at the turbine inlet–specifically, the number of hours that the engine is operated above base load and peaking temperature limits.

When a gas turbine is used exclusively for very short-term duty, such as for providing emergency or peaking power, a higher temperature limit, referred to as a "peak load" rating, may be permitted in recognition of the application. This higher peak load must be reflected in shorter inspection intervals.

Environment

If the engine is in an environment that results in the ingesting of abrasive or corrosive elements, it can have a significant influence on maintenance requirements. The use of proper materials and protective coatings and the incorporation of inlet air filtering and washing equipment are steps that can be taken to minimize the effects of environment on maintenance.

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APPENDIX C

TURBO POWER & MARINE MAINTENANCE PROCEDURES

The enclosed material was provided by Turbo Power & Marine for use in this study, and it discusses maintenance procedures and recommendations for Turbo Power & Marine gas turbine units.

MAINTENANCE

Inspection Intervals

INSPECTION INTERVALS The following nominal inspection intervals are recommended for the TP&M gas generators and/or gas turbines:

Type of Servicing	Approximate Man Hours to Accomplish	Frequency
Routine Inspections		
Visual Inspection of Engine	1/2 Man-Hour	Every 50 Hours
Oil Sample	10 Minutes	Every 300 Hours
Oil Change and Visual Inspection of Engine	4 Man-Hours	Whenever Sample Analysis Indicates Need.
Hot Section Inspections		
Initial Hot Section Inspection of Gas Generator	24 Man-Hours (3 Men)	25% of Nominal Interval Given in Table I
Subsequent Hot Section Inspection of Gas Generator	24 Man-Hours (3 Men)	Interval to be determined by Results of Initial Inspection and Operating Experience. Nominal Intervals Given in Table I or Annual, Whichever Occurs First.
Major Field Inspections	60 Man-Hours Depending on Nature of Work (3 Men)	Determined by Results of Hot Inspection Section. Nominal Intervals are Given in Table IA.

RECOMMENDED HOT SECTION INSPECTION INTERVALS (HOURS)

Hot Section Inspection (Hours)

Rating	Rating Factor	Natural Gas TPM-FR-2	MIL-J-5624 Jet A Kerosene TPM-FR-1	MIL-F-16884 No. 2 Burner - Diesel Heavy Distillate TPM-FR-1
Base Load	1.00	3,750	3,000	2,600
Peak or Max. Continuous	2.25	1,660	1,335	1,165
Reserve Peak	9.0	415	330	290
Fuel Factor		1	1.25	1.43

Note:

1. Turbo Power and Marine Systems, Inc. should be consulted for recommendations if the engine has been subjected to 14.0 hours of continuous operation at maximum capability or equivalent rating.

RECOMMENDED MAJOR FIELD INSPECTION

Major Field Inspection (Hours)

Rating	Rating Factor	Natural Gas TPM-FR-2	MIL-J-5624 Jet A Kerosene TPM-FR-1	MIL-F-16884 No. 2 Burner - Diesel Heavy Distillate TPM-FR-1
Base Load	1.0	7,500	6,000	5,250
Peak or Max. Continuous	1.5	5,000	4,000	3,250
Reserve Peak	6.0	1,250	1,000	875
Fuel Factor		1	1.25	1.43

Note:

1. Turbo Power and Marine Systems, Inc. should be consulted for recommendations if the engine has been subjected to 14.0 hours of continuous operation at maximum capability or equivalent rating.

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ROUTINE, HOT SECTION AND MAJOR FIELD INSPECTIONS

Component	Nature of Inspection	Routine	HSI	MFI
Gas Generator General				
External Tubing, Hose and Electrical Leads	Security of all connections, clamps and brackets	x	x	x
	Evidence of chafing, wear and cracks	x	x	x
	Evidence of fuel, gas, air or oil leak	x	x	x
Mounts (See Aux. Equip. List)	Security	x	x	x
	Lubrication	x	x	x
External Case Bolts	Security of lockwire bolts and nuts and missing bolts and nuts	x	x	x
Low Compressor				
Inlet Guide Vanes	Nicks, dents, cracks, evidence of FOD			x
Nose Cone	Security, dents and cracks			x
Bellmouth	Check clearance and security			x
1st-4th Stage disk and blade assembly	Nicks, dents, cracks, erosion, corrosion, evidence of DOD/FOD			x
	Blade and Spacer freedom			x
Stator assemblies (1st-3rd)	Nicks, dents, cracks (Including inner & outer shrouds)			x
	FOD/DOD			x
	Erosion, corrosion			x

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Component	Nature of Inspection	Routine	HSI	MFI
Disk & Blade Assemblies 3rd-8th (thru Bore-scope Holes)	Nicks, dents, cracks, erosion			x
	FOD/DOD			x
Low Compressor General	Contamination			x
High Compressor				
9th Stage Disk & Blade Assembly (through Bore-scope Hole)	Nicks, dents, cracks, erosion, missing shrouds			x
	Evidence of FOD/DOD			x
15th Stage Stator Assembly	Nicks, dents, cracks, erosion			x
	DOD/FOD			x
High Compressor General	Contamination			x
Combustion Section				
Fuel Manifold & Nozzle Assembly	Carbon Build-up		x	x
	Loose Nozzle Nuts		x	x
	Wear on Nozzle nuts		x	x
	Clogged Nozzles		x	x
	Fuel Leakage		x	x
	Nicks and Dents		x	x
	Distortion		x	x
Combustion Chambers	Cracks, Warpage		x	x
	Localized overheating		x	x
	Burning, crossover tubes, liners		x	x
	Locating lug & swirl vane wear		x	x

Component	Nature of Inspection	Routine	HSI	MFI
Combustion Chamber Positioning Pin	Wear, Fit		x	x
	Cracks		x	x
Combustion Chamber Clamp	Wear, Fit,		x	x
	Cracks		x	x
Outer Combustion Chamber Cover	Cracks, Wear at combustion chamber positioning pin location		x	x
	Drain Valve operation		x	x
	Cracks		x	x
Transition Duct	Cracks, localized burning, warping		x	x
Turbine				
First Stage Nozzle Guide Vane (NGV)	Bowing, coating deterioration		x	x
	Nicks, dents, cracks, erosion		x	x
	Plugged cooling air holes		x	x
First Stage Turbine	Nicks, dents, cracks, erosion		x	x
	Coating deterioration, "Hot Corrosion"		x	x
	Blade creep			x

Component	Nature of Inspection	Routine	HSI	MFI
First Stage Outer Air Seal	Erosion, corrosion of knife-edge seals		x	x
	Rolled or damaged knife-edge seals		x	x
First Stage NGV Retaining Ring	Wear, cracks		x	x
2nd Stage NGV	Nicks, dents, cracks		x	x
	Erosion, coating deterioration, "Hot corrosion"		x	x
Turbine Exhaust Case				
Exhaust Pressure Probes (PT7)	Security	x	x	x
	Cracked probes or bosses	x	x	x
Exhaust Gas Temp Probes (Tt7)	Broken terminals	x	x	x
	Security	x	x	x
	Chaffed or broken wires	x	x	x
Oil System				
Strainers	Check for metal and foreign particles	x	x	x
	#6 Bearing (external)	x	x	x
Oil Drain Plugs	Check for metal particles	x	x	x
Fuel System				
Fuel Pump	Check for leakage	x	x	x
	Filter for damage and contamination	x	x	x
	Security	x	x	x

Component	Nature of Inspection	Routine	HSI	MFI
Throttle Valve & Actuator	All connections for wear and damage	x	x	x
	Security	x	x	x
	Leakage	x	x	x
	Filters for damage and contamination	x	x	x
Pressurizing & Dump valve	Security of lines, fittings and mounting	x	x	x
	Screens for damage and contamination	x	x	x
	Leakage	x	x	x
Ignition System				
Wiring	See electrical leads	x	x	x
Exciters	Sécurity of mounting	x	x	x
Igniters	Electrodes & Ceramic insulation for cracks & erosion		x	
	Check for operation	x	x	x
Anti-Icing System				
Anti-Icing Valve	Security	x	x	x
	Operation	x	x	x
	Electrical Leads (See wiring)	x	x	x
Free Turbine				
Flexible coupling & Free Turbine Output shaft	Check for nicks, dents and cracks	x	x	x

Component	Nature of Inspection	Routine	HSI	MFI
Tubes (external)	Check for nicks, dents and chafing	x	x	x
	Scratching Security of clamps and brackets	x	x	x
	Evidence of air or oil leaks	x	x	x
Oil System				
Strainers	Check for metal and foreign particles	x	x	x
Free Turbine oil tank	Check for crack	x	x	x
	Bulge or buckling	x	x	x
External case bolts	Check for security, missing bolts & nuts	x	x	x
Electrical Leads	Check for chafing, broken insulation, security	x	x	x

AUXILIARY EQUIPMENT

Equipment by TPM or to TPM Specification

Item	Hours	Weeks	Months	Years	Maintenance
Throttle Valve Strainers	100				Remove & clean main & servo sensor. (Recommend sonic cleaning)
P&D Valve Strainer	100				Remove & clean (same as above)
Fuel control					
Chassis slides			6		Graphite powder.
Visual inspection			6		
Basic checkout of control	8000 or			1	(whichever is first)
N_3 overspeed check	400 or		6		(whichever is first) check to specified limits and adjust if necessary. (operation at overspeed held to minimum)
Sequencer					
.Visual Inspection				1	Inspect for broken wires, loose connections. (Insure all station power AC & DC to sequencer is off.
Cabinet Fans				1	Clean blades.
Air Filters				1	(or sooner if dirty) replace or clean filter elements

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Item	Hours	Weeks	Months	Years	Maintenance
Speed Relays N_3 overspeed check	400 or		6	1	Check speed settings with counter, (whichever is first) check to specified limits and adjust if necessary. (operation at overspeed held to minimum)
Shut-off Valves (2) Redundancy	400 or		6		(whichever is first) check that either valve will shut down unit.
GG & FT Lube Oil System Oil System Filters (Cuno)	100 or		1		Remove cover & clean every month or 100 hrs, whichever occurs first. Clean at each oil change.
GG & FT Mount System		1			Check for clearance in plenum opening. Check for security.
Front Mount		1			Check for security. Grease fittings.
Center Ball Joint		1			Grease
Keel Mount		1			Grease fittings

Item	Hours	Weeks	Months	Years	Maintenance
Gas Generator Inlet & Plenum Chamber Plenum for loose objects and cleanliness			1		Inspect-clean-repair or secure as necessary.
Gas Turbine Enclosure Secondary Air inlet doors			6		Exercise doors up & down three or four times. Correct if necessary to ensure free movement.
Floor		1			Visual inspection for fuel or lube oil accumulation including generator area behind free turbine elbow.
Hoses and electrical cable			1		Visual check for leaks, looseness, chafing, damage etc. Correct as necessary.
Interior of Exhaust Stacks	150				Visual check-flashlight looking down stack & through access hatch in rear of stack; check for: 1) Sheet failures, cracks, etc. 2) Groups of weld failures.

Item	Hours	Weeks	Months	Years	Maintenance
Interior of Exhaust Stacks (Cont'd)					3) Bulges in perforated sheets greater than 38.1mm inch deep. 4) Corrosion 5) Baffles devoid of insulation
Turning Vanes			6		Visual check for looseness of bolts excessive wear and fatigue, cracking of vanes or gusset plate.
Fire Protection System CO_2 fire extinguishing equipment control cylinder		1			Check control system pressure. The low side gage on regulator to read 290 to 310 psig. (1999 to 2137 kPa)
Protect-A-Fire				1	Visual inspection for dust accumulation.
Main Cylinder Bank				1	Field test according to operating instructions.

Item	Hours	Weeks	Months	Years	Maintenance
Secondary Air Damper Doors			6		or after painting damper doors. Disconnect cable ring. Close doors. Reset doors in open position. Lubricate pinnions with light oil if necessary.
Fenwall Halon fire suppression system					
Fire Detector				1	Visual inspection for dust accumulation.
Halon Containers			3		Check pressure. Check hardware for looseness or damage.
			6		Remove and weigh containers.
Control System			3		Check circuit integrity with "Push-To-Test" button
				1	Service initiators and connections per vendors manual.
				2	Replace initiators

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Item	Hours	Weeks	Months	Years	Maintenance
Secondary Air Damper Doors			6		Same as above in CO_2 system.
Temperature Instruments L&N Recorder Ink Pad		6			Check ink supply.
Print wheel		6			Correct faulty printout.
Slide Wire			2		Clean.
Printing Carriage Shaft			2		Clean & Oil.
Gear & clutch assembly			2		Clean & Lube.
Electrolytic Capacitors			6		Replace.
Vibration Monitor Reliance				1	Calibrate in accordance with instructions.
Ice Detector		1		1	Self test operation, calibrate.
Chip Detector			6		Integrity test.
Starting Air Supply System Compressor Crankcase		1			Check oil level, add as required.
Oil Change	300 or		3		Drain & Replace oil every three months or 300 hours, whichever occurs first.

Item	Hours	Weeks	Months	Years	Maintenance
Cooling			3		Clean inter-cooler fins and cylinders.
Intake Filter			1		Clean or re-place filter element.
Stage Pressures			1		Check 1st and 2nd stage pressures.
Condensate Draining			1		Check opera-tion of drain & timer.
Drive			1		Check belt tension.
Mounting			3		Check for ex-cessive vibration.
Intake Air Shutter			1		Check that shutter opens and closes with thermostat.
Exhaust Fan Operation and Lubrication			1		Check as above, oil fan motor.
Compressed Air Filter			6		Check pressure drop across filter.
DC Motor				1	Lubricate.
AC Motor			1		Lubricate.

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Item	Hours	Weeks	Months	Years	Maintenance
Soft Start Control Valve Strainer-Signal Pressure Line					
Electric Generator Electric Machinery 25 MVA					
Oil Filter Element			6		Replace filter element initially at 6 months, annually thereafter.
Stator Windings			6		Insulation Resistance Test.
Rotor Windings			6		Insulation Resistance Test. Inspect windings for oil leak contamination.
Air Filters			3		Remove & clean, replace if clogged or damaged.
Brushless Exciter			3		Inspect exciter rotor stator & diode wheel.
			3		Check for secure mounting of diodes on heat sinks.

Item	Hours	Weeks	Months	Years	Maintenance
Brushless Exciter (Cont'd)			3		Inspect fuses for open or short circuits.
Condensate Collector Filter Element			1		Replace element.
Leak check		1			Listen for leaks with unit off.
Leak check			1		Check pressure piping with soapy water.
Control Valve			1		Listen for leak past valve seat.
Pressure Vessels			1		Drain condensate through manual valve.
Building Heater			6		Lower thermostat to check operation.
Compressor Unloading Drain Valve			1		Listen for leak in drain line with unit on.
Discharge line check valve			1		Listen for leak with unit off.

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Item	Hours	Weeks	Months	Years	Maintenance
Strainer-Signal Pressure line		1			Open drain petcock to exhaust any accumulated moisture from filter housing.
Liquid Fuel Forwarding System Filter/Separator	daily with experience indicates proper interval				Check differential pressure gage. Periodically note liquid level in sight gage.
Liquid Fuel Final Filter	weekly until experience indicates proper interval				Check pressure drop across filter. Replace element when Δ D = 15 psig (103 kPa) Replace Buna N Gasket
AC & DC Fuel Pumps			3		Inspect for loose bolts, excessive vibration, rust and corrosion.

Included in the price are special hand tools, adapters, pullers, etc. required for a hot section inspection and light maintenance. Tools such as slings, adapters and saddles required for removing a gas generator or free turbine are available on a no-charge loan basis.

APPENDIX D

OPERATION AND MAINTENANCE COSTS EXPERIENCE

The data presented in Tables D.1 through D.3 are abstracted from "Gas Turbine International."[16] They summarize the results of an annual national survey regarding operation and maintenance costs for electric utility gas turbines.

TABLE D.1

TURBO POWER AND MARINE TURBINE COST DATA

Maintenance Costs (Dollars Per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars)
4	19.0	1,558	265	272	7
255	48.6	1,699	686	1,120	434
1	16.3	1,409	240	242	2
18	17.1	1,424	248	273	25
14	22.1	702	311	321	10
259	230.2	2,938	2,996	3,757	761
20	16.0	958	466	485	19
37	29.7	2,268	374	459	85
77	32.0	1,697	451	581	130
93	20.5	1,111	231	334	103
32	26.6	1,568	350	400	50
24	7.7	633	122	127	15
88	21.5	1,245	299	408	109
4	15.2	2,348	228	237	9
20	22.1	1,641	304	337	33
67	12.3	611	163	204	41
36	25.3	1,344	241	290	49
250	46.5	3,096	676	1,451	775
219	14.6	963	211	422	211
11	14.0	470	207	212	5
36	8.3	587	90	111	21
459	19.2	653	270	570	300
375	95.0	99.8	1,401	1,775	374
21	10.8	1,003	165	186	21
208	10.3	693	129	273	144
28	18.1	1,134	214	246	32

TABLE D:1 (continued)

Maintenance Costs (Dollars Per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars)
108	46.2	1,117	604	725	121
241	61.2	1,226	768	1,063	295
327	69.1	1,642	891	1,428	537
227	74.4	1,545	899	1,250	351
210	51.4	1,173	623	869	246
76	39.6	1,210	497	589	92
356	61.5	1,125	776	1,176	400
107	40.2	1,519	562	724	162
184	48.8	1,493	593	867	274
212	42.3	1,397	479	775	296
199	13.9	1,049	158	367	209
169	416.6	2,593	5,241	5,678	437
469	356.8	1,831	4,455	5,313	858
124	15.6	976	188	309	121
24	9.0	638	123	138	15
110	3.5	264	46	75	29
121	15.0	1,036	187	312	125
27	28.4	1,798	257	305	48
76	17.9	1,178	225	315	90
152	17.1	1,064	216	478	162
60	37.7	604	393	429	36
497	4.9	396	72	269	197
135	2.8	311	43	85	42
18	24.2	1,684	376	406	30
18	27.6	1,756	282	314	32
88	27.9	1,772	4-0	556	156
43	11.5	749	206	238	32

TABLE D.1 (continued)

Maintenance Costs (Dollars Per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars	Maintenance Costs (Units of Thousands of Dollars
22	6.4	556	117	129	12
88	32.0	970	294	379	85
52	35.0	1,164	385	445	60
5	29.1	1,712	484	492	8
276	117.5	1,854	868	1,380	512
73	41.2	1,293	576	670	94
132	118.4	2,309	889	1,193	304
244	180.7	1,933	3,652	4,124	472
47	6.0	404	112	131	19
62	212.7	1,527	3,305	3,399	94
31	74.2	1,078	738	771	33
63	78.9	2,058	626	755	129
31	138.1	3,994	1,215	1,339	124
176	179.8	2,663	1,921	2,391	470
106	1.8	47	24	29	5
TOTAL →	3,635.9	91,460	46,122	58,172	12,050
8,442					

TABLE D.2

WESTINGHOUSE TURBINE COST DATA

Maintenance Costs (Dollars Per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars)
309	16.8	863	284	551	267
68	22.8	893	244	305	61
365	20.2	356	358	488	130
1,161	1.0	68	18	97	79
944	1.3	108	22	124	102
455	1.2	90	25	66	41
480	9.7	206	93	192	99
34	88.1	1,435	1,251	1,300	49
31	105.4	1,972	1,339	1,401	62
307	11.0	630	187	381	194
815	2.9	233	29	219	190
151	13.4	969	127	274	147
523	.15	21	3	14	11
33	8.7	720	90	114	24
48	38.0	889	362	405	43
35	66.1	1,723	1,267	1,329	62
101	44.7	1,516	1,052	1,205	153
222	45.3	1,824	448	853	405
82	296.1	1,863	6,301	6,453	152
15	54.8	2,406	780	817	37
1,124	3.1	177	42	241	199
79	13.3	750	139	198	59
8	46.0	2,135	430	447	17
12	41.4	1,727	200	221	21
470	2.1	149	43	113	70

TABLE D.2 (continued)

Maintenance Costs (Dollars Per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars)
9	12.4	976	75	84	9
18	46.8	820	216	231	15
18	37.0	978	148	166	18
36	1.1	110	5	9	4
1,316	1.2	117	6	160	154
1,063	2.5	112	11	130	119
80	1.1	100	7	15	8
5	42.4	3,775	337	335	18
18	116.2	2,315	724	766	42
28	78.0	1,553	490	534	44
73	89.9	1,943	568	709	141
7	122.6	8,760	1,063	1,120	57
16	148.9	8,760	1,185	1,321	136
23	5.3	432	92	102	10
36	81.5	4,526	1,163	1,326	163
71	227.2	2,199	2,632	2,788	156
9	278.0	2,221	3,014	3,033	19
60	34.8	616	361	398	37
111	48.3	872	569	666	97
14	14.4	1,199	226	243	17
90	184.3	2,546	2,299	2,528	229
8	34.6	1,548	470	483	13
101	1.6	119	9	21	12
34	58.0	2,139	456	528	72
65	50.3	1,136	282	356	74
TOTAL →	2,671.95	73,595	31,586	36,230	4,644
11,181					

TABLE D.3

GENERAL ELECTRIC TURBINE COST DATA

Maintenance Costs (Dollars per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars)
71	28.8	1,137	404	485	81
14	14.3	1,226	193	210	17
47	61.7	1,867	727	815	88
26	18.9	1,509	299	338	39
8	16.5	1,210	224	234	10
36	72.9	1,684	1,135	1,196	61
14	112.9	8,784	1,512	1,635	123
16	2.0	250	35	39	4
20	1.9	204	22	26	4
49	22.7	1,601	377	455	78
578	1,636.0	4,163	20,125	22,531	2,406
116	629.8	2,199	7,981	8,236	255
686	1,049.0	3,998	11,935	14,676	2,741
96	47.4	739	537	608	71
129	5.4	831	86	193	107
200	32.5	594	363	482	119
15	26.5	1,880	347	376	29
70	381.6	3,494	2,517	2,762	245
50	109.3	1,933	1,423	1,520	97
24	28.2	2,007	356	405	49
16	30.2	1,865	409	439	30
37	92.6	2,234	1,243	1,326	83
29	31.5	920	400	427	27
26	25.9	1,589	347	389	42
49	77.3	2,955	972	1,117	145
75	12.3	574	163	206	43
26	63.1	2,578	780	846	66

TABLE D.3 (continued)

Maintenance Costs (Dollars Per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars)
35	27.4	775	400	427	27
22	63.4	2,981	584	649	65
254	76.5	1,540	1,088	1,479	391
107	83.8	1,559	1,146	1,313	167
30	25.6	967	424	453	29
20	40.2	1,320	569	696	26
41	179.5	2,512	2,273	2,376	103
123	7.5	309	109	147	38
10	58.7	1,541	805	820	15
27	36.8	1,396	555	593	38
38	14.2	651	214	239	25
5	23.7	841	322	326	4
33	3.7	398	55	68	13
13	79.3	3,227	637	680	43
41	161.9	1,515	1,691	1,753	62
741	5.4	212	27	184	157
92	2.9	195	15	33	18
28	114.5	2,549	880	952	72
56	14.1	1,101	135	197	62
299	86.4	796	1,380	1,618	238
74	126.5	1,319	1,505	1,602	97
96	2.3	187	39	57	18
11	85.3	3,046	941	975	34
39	68.1	1,368	986	1,039	53
39	61.7	1,269	887	936	49
46	98.5	1,620	1,467	1,541	74
23	138.2	8,784	1,171	1,370	199
20	115.7	7,272	1,266	1,415	149
109	22.6	387	380	422	42
66	23.7	422	420	448	28

TABLE D.3 (continued)

Maintenance Costs (Dollars per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Total (Units of Thousands of Dollars)	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars
73	560.9	1,949	9,651	9,794	143
15	56.9	1,873	784	812	28
62	353.2	2,264	3,855	3,996	141
9	15.8	1,107	240	250	10
9	14.3	1,110	255	265	10
36	32.9	1,268	681	727	46
45	8.9	926	156	198	42
18	12.0	1,073	178	197	19
87	2.4	207	35	53	18
388	2.3	201	35	113	78
12	30.1	2,343	315	342	27
5	45.8	2,306	283	294	11
22	46.4	2,925	319	384	65
66	24.3	865	247	304	57
5	25.7	1,864	180	190	10
147	259.9	3,357	3,359	3,852	493
541	325.9	20,060	5,950	7,065	1,115
191	266.0	2,209	4,477	4,899	422
181	420.7	3,117	5,797	6,360	563
444	137.2	1,775	2,260	3,048	788
245	124.4	1,628	2,310	2,709	399
64	124.9	2,689	883	1,056	173
236	312.6	1,908	3,833	4,283	450
87	132.0	2,956	1,094	1,350	256
96	23.4	407	428	467	39

TABLE D.3 (continued)

Maintenance Costs (Dollars per Hour of Operation)	Net Energy Generation (Millions of KW-HR)	Hours Run - Gas Turbine Connected to Load	Fuel (Units of Thousands of Dollars	Total (Units of Thousands of Dollars)	Maintenance Costs (Units of Thousands of Dollars)
100	28.2	482	386	434	48
28	30.4	542	223	238	15
45	12.7	538	113	137	24
566	1.5	83	19	66	47
322	20.8	518	339	506	167
13	2.5	236	43	46	3
133	15.1	345	144	190	46
48	5.1	331	34	50	16
15	111.7	5,389	839	922	83
28	47.4	977	560	587	27
6	64.8	4,490	334	361	27
130	6.3	108	36	50	14
23	67.2	6,328	576	722	146
141	2.6	241	51	85	34
24	17.0	1,395	149	182	33
139	7.5	553	63	140	77
7	106.8	7,998	520	577	57
39	5.1	310	77	89	12
21	55.5	2,754	465	522	57
5	89.0	5,352	457	485	28
TOTAL →	10,626.4	190,346	131,642	147,673	16,031
.0,277					

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